DOE/NASA CONTRACTOR REPORT

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SOLAR HEATING AND COOLING SYSTEM DESIGN AND DEVELOPMENT - FINAL REPORT

Prepared by

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P. O. Box 8661
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Under Contract NAS8-32092 with

National Aeronautics and Space Administration
George C. Marshall Space Flight Center, Alabama

For the U. S. Department of Energy

U.S. Department of Energy

Solar Energy

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SECTION I

PROGRAM PLAN

1.1 SCOPE

This project, a part of the Marshall Space Flight Center program for the development of solar heating and combined solar heating and cooling systems(1), involved the complete design and development of marketable systems for single family and commercial applications and the delivery, installation, and monitoring of the prototype systems. The development of the two types of systems proceeded in parallel with selected commonality of system elements. The time required for the development of the combined heating and cooling systems was greater than for the heating systems, so the heating systems were being installed during the development of the cooling subsystem.

1.2 SCHEDULE

A summary program schedule is shown in Figure 1-1.

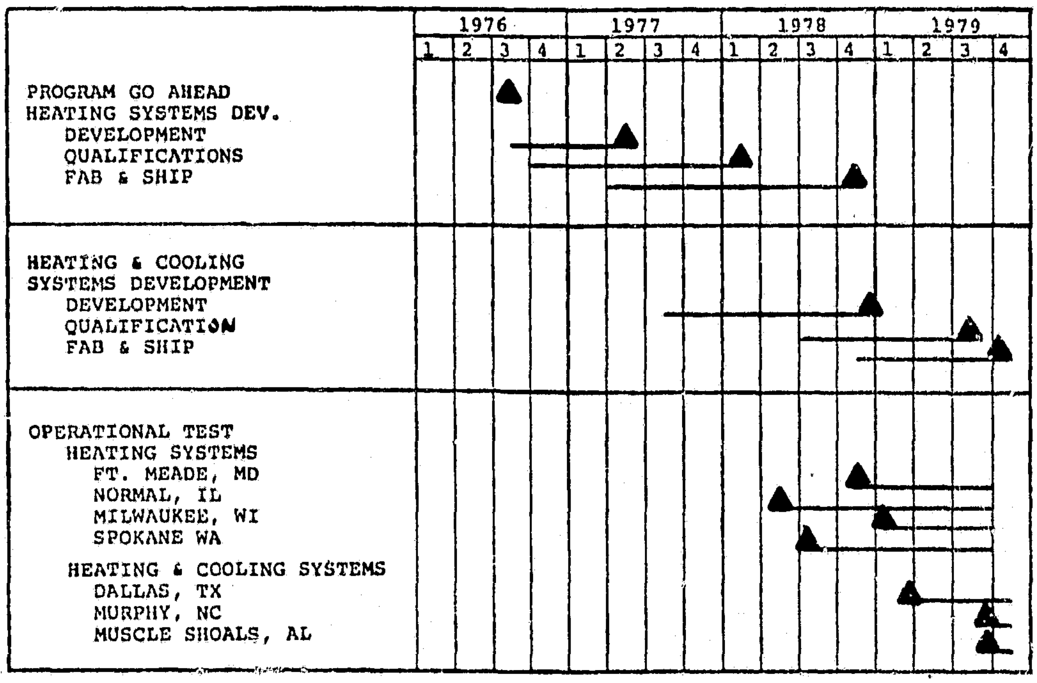


FIGURE 1-1 Summary Program Schedule

(1) This program is a part of the Department of Energy's activity to develop and demonstrate solar heating and combined heating and cooling systems.

The summary program schedule in Figure 1.1 is a schedule which evolved over the course of the program. The program which began in 1976 underwent a major redirection in mid-1977 to direct the cooling systems designs away from the proposed systems to units with higher performance. The resultant stretching of the technology increased the amount of development required, particularly in the solar collector and Rankine electric heat pump programs. The systems which are described herein are those resulting from the redirection. A secondary result of the increased development was a delay in delivery and installation of the cooling systems into 1979.

1.3 Cost

The projected cost of contract NAS8-32092 for the Solar Heating and Cooling System Design and Development is $8,591,800. This value includes a General Electric Co. direct contribution of $1,287,400. The costs include design, development, test and installation with the exception of the two TVA sites (Murphy, N.C. and Muscle Shoals, Ala.) where TVA is responsible for the installation. The costs of installation were not a part of the original bid, but were incorporated into the overall program in a modification to the contract negotiated over a period of time as sites were selected and approved.

As of October 26, 1979, all of the systems have been delivered and installed except for the two TVA sites.

SECTION 2

SYSTEM DESCRIPTION

2.1 SOLAR HEATING AND HOT WATER SYSTEMS

Application surveys and performance studies were conducted to arrive at a solar heating hot water configuration that could be used in a wide variety of applications. The goal was to identify subsystem modules that could be utilized in building-block fashion to adapt standard hardware items to a variety of applications. Typical subsystems are shown in Figure 2-1 which includes the cooling subsystem. The family of solar heating systems is shown in Figure 2-2 for three types of applications. The configurations are similar, with the commercial and multi-family systems involving multiple zones and a central boiler for auxiliary heat. Note that the residential system is compatible with a heat pump installation. A system test to verify the performance of this configuration has been completed at the General Electric Valley Forge test facility and the configuration is being used for the four solar heating and hot water test installations.

Features of the heating system include:

- Control of solar collector loop operation based on average insolation level to minimize parasitic power and number of pump starts and maximize amount of energy collected.
- Prepackaged control and auxiliaries (pumps and heat exchangers) for single family residences.
- Families of equipments produced to General Electric specifications.

- Control of energy flow to maximize use of solar energy and minimize use of auxiliary energy.
- Compatibility with many types of existing HVAC equipment to minimize the retrofit problem.

The collector loop design has progressed from one which contained an overtemperature heat exchanger (heat dump) which operated to keep the collectors below 320°F to a simplified approach that eliminated the heat dump and also incorporates mechanical valves to maintain a minimum operating pressure level. The initial collector loop design is shown in Figure 2-3 while the updated design is shown in Figure 2-4. The early heating and hot water operational test sites have the first loop but the heating, cooling, and hot water sites have the updated design.

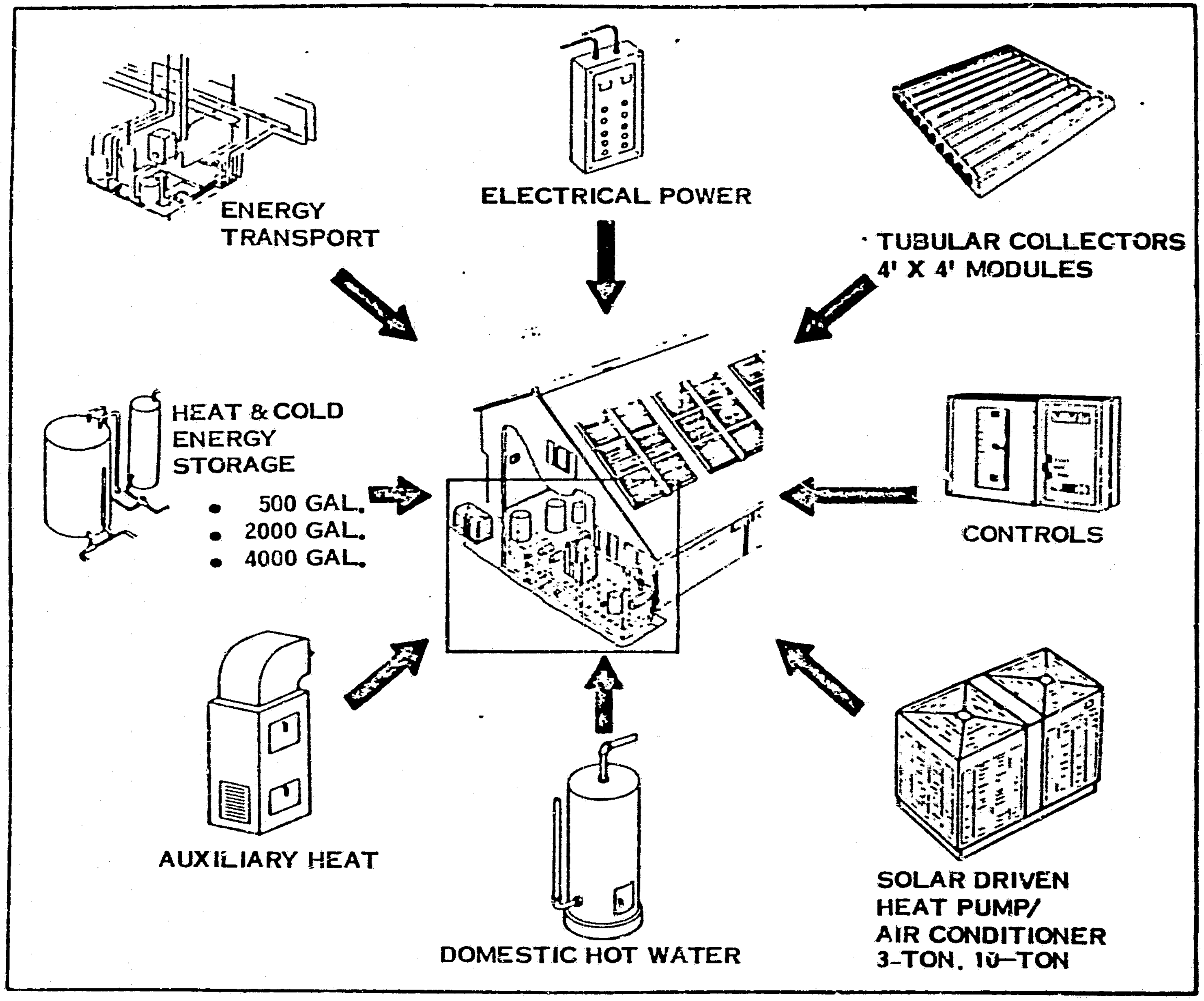


FIGURE 2-1 Subsystem Building Blocks

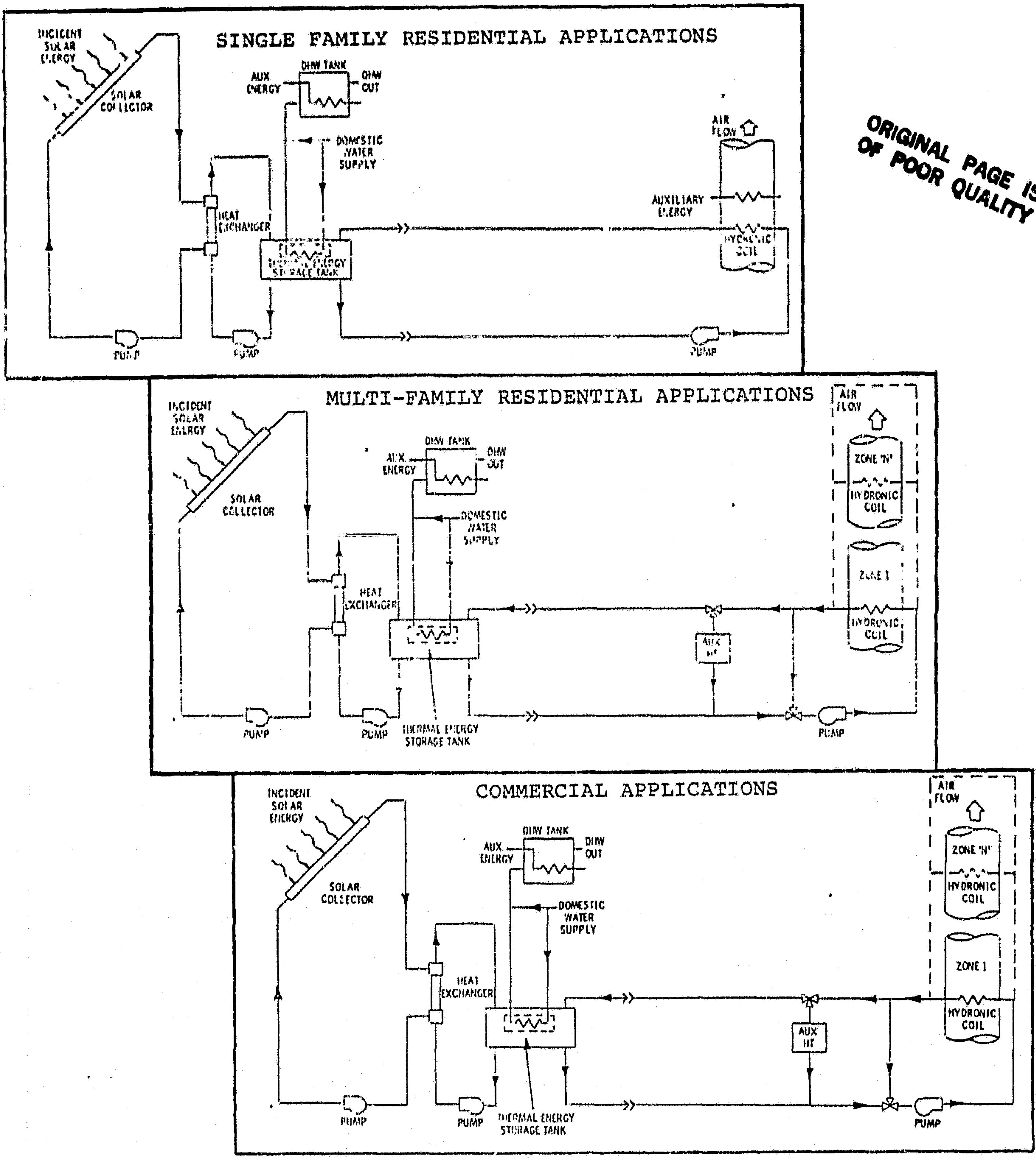


FIGURE 2-2 Solar Heating System Configurations

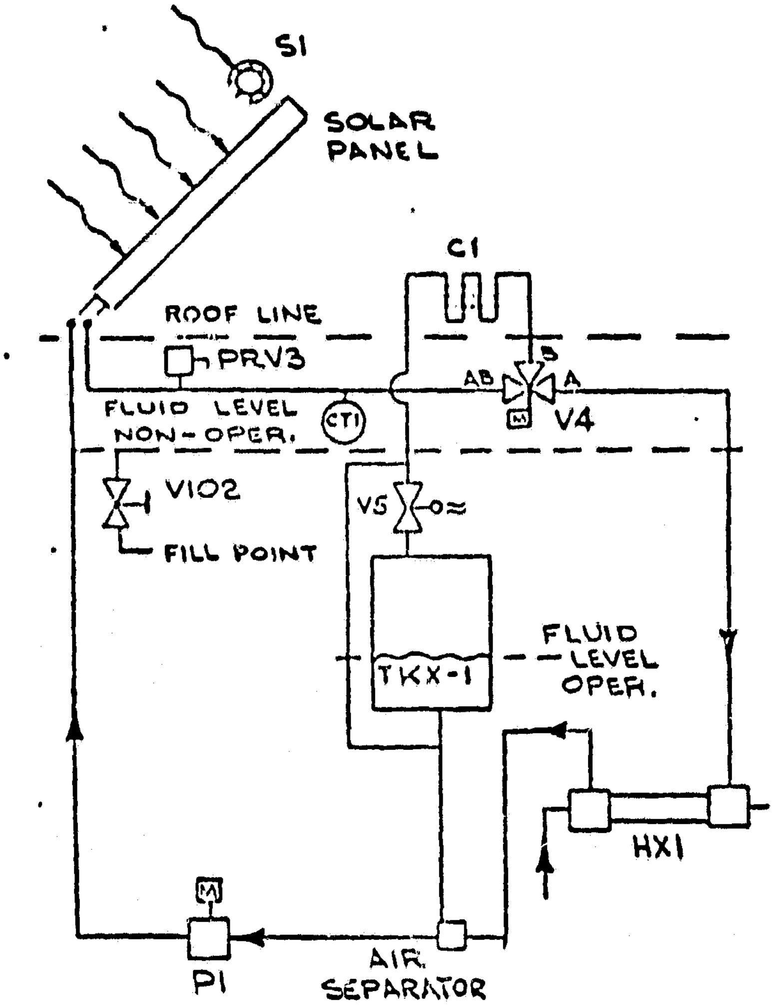


FIGURE 2-3 Solar Collector Loop I

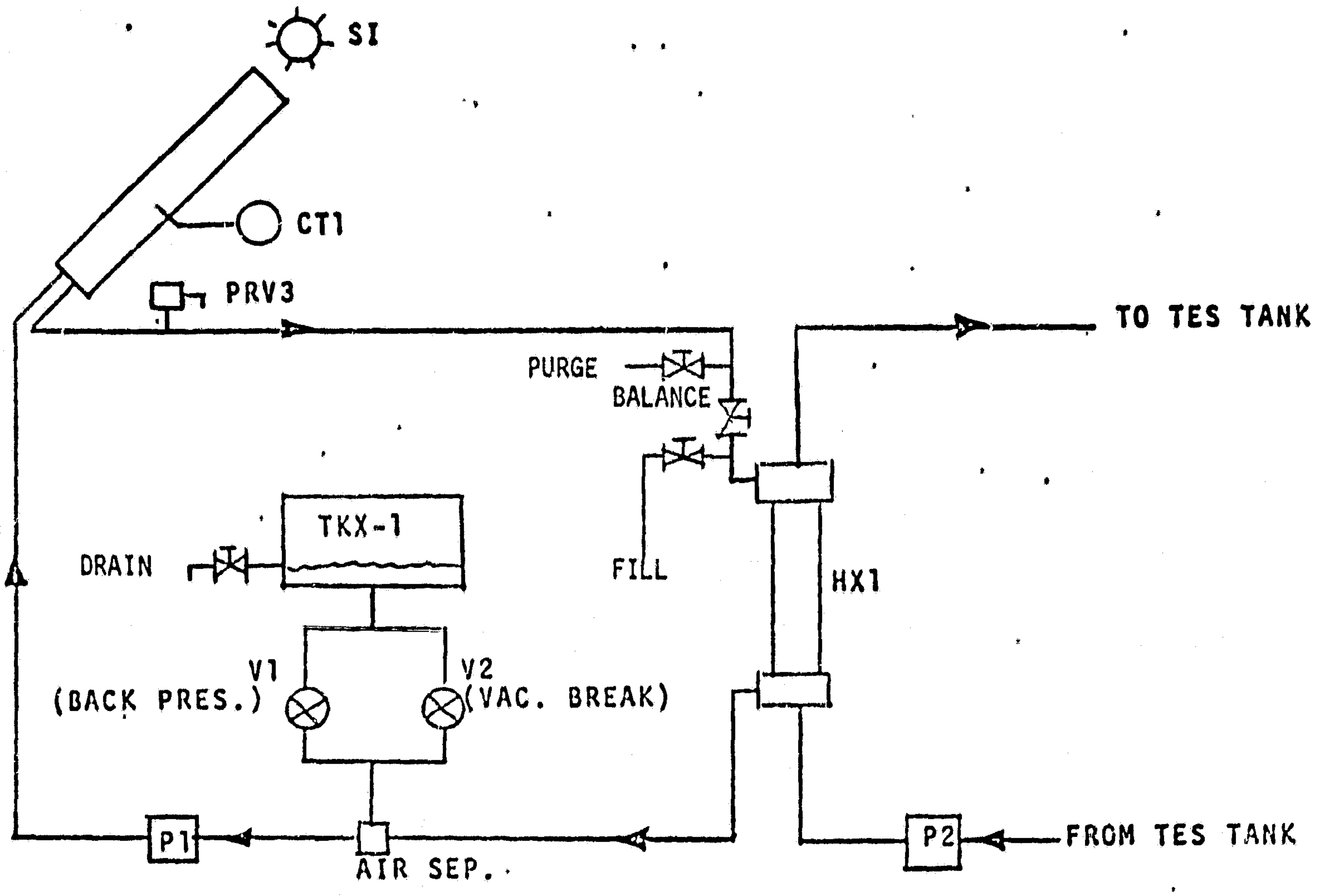


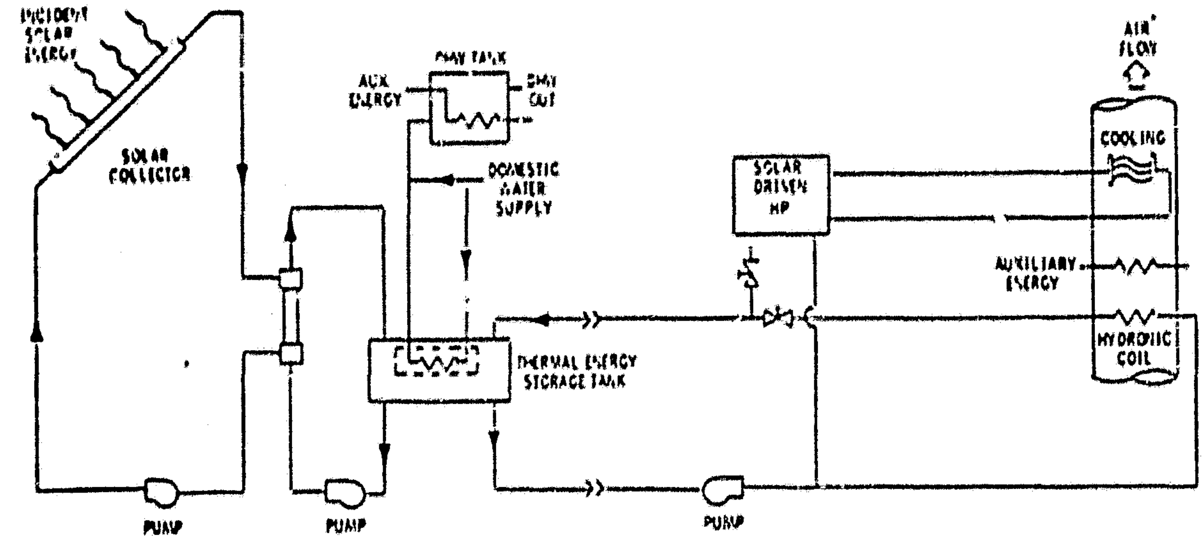
FIGURE 2-4 Solar Collector Loop II

2.2 SOLAR HEATING, COOLING AND HOT WATER SYSTEMS

Heating and cooling system configuration studies to efficiently utilize solar energy and be compatible with the heating only system configuration and equipment were carried out. The selected configurations are shown in Figures 2-5 and 2-6. The single family and the commercial systems selected are similar and strongly resemble the heating system configurations. Systems using heat pumps are illustrated, but the configuration is applicable to systems with air conditioners. For the multi-family applications involving a central cooling unit and multiple zones, it was found to be more effective to use a chilled water distribution system as shown in Figure 2-6.

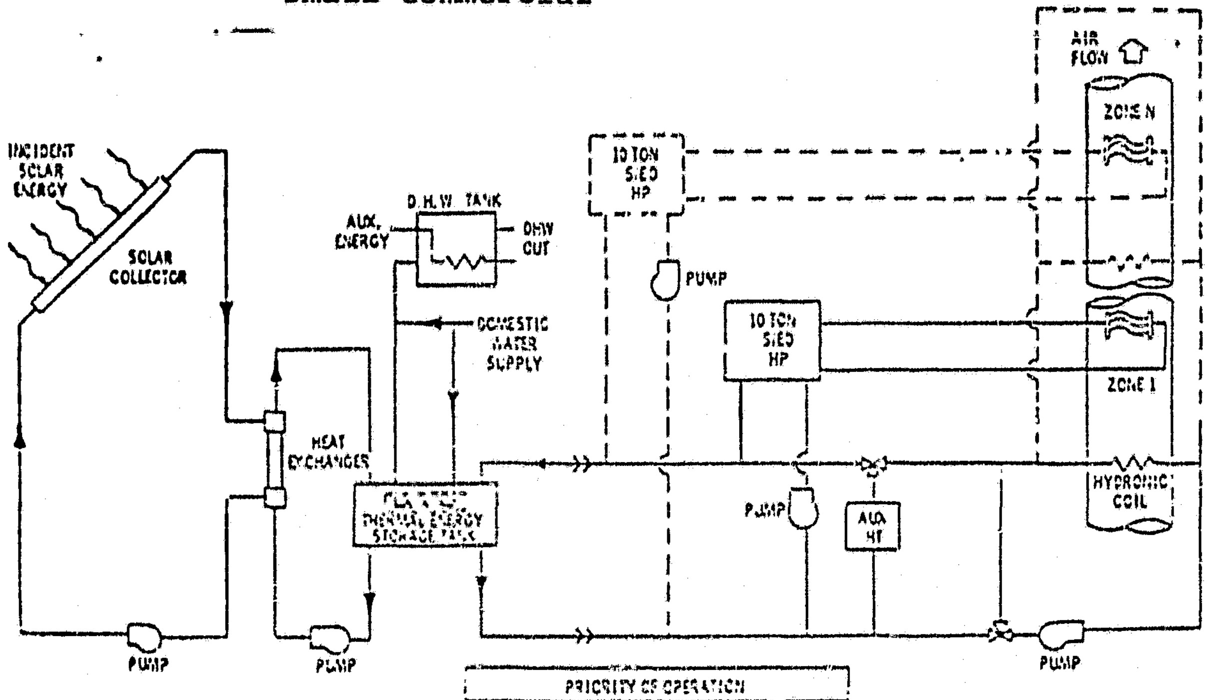
Design studies have been carried out to pre-engineer systems of various sizes to minimize the application engineering needed for a variety of installations.

As a result of market studies a 3-ton heat pump size was selected for single-family residential applications and a 10-ton size for commerical and multi-family residential applications. These are regarded as market entry units and other sizes will be needed later to fill out the product line.



PRIORITY OF OPERATION	
HEATING	COOLING
1. DIRECT SOLAR 2. ELECTRIC HEAT PUMP 3. AUXILIARY ENERGY	1. SOLAR DRIVEN 2. ELECTRIC DRIVEN

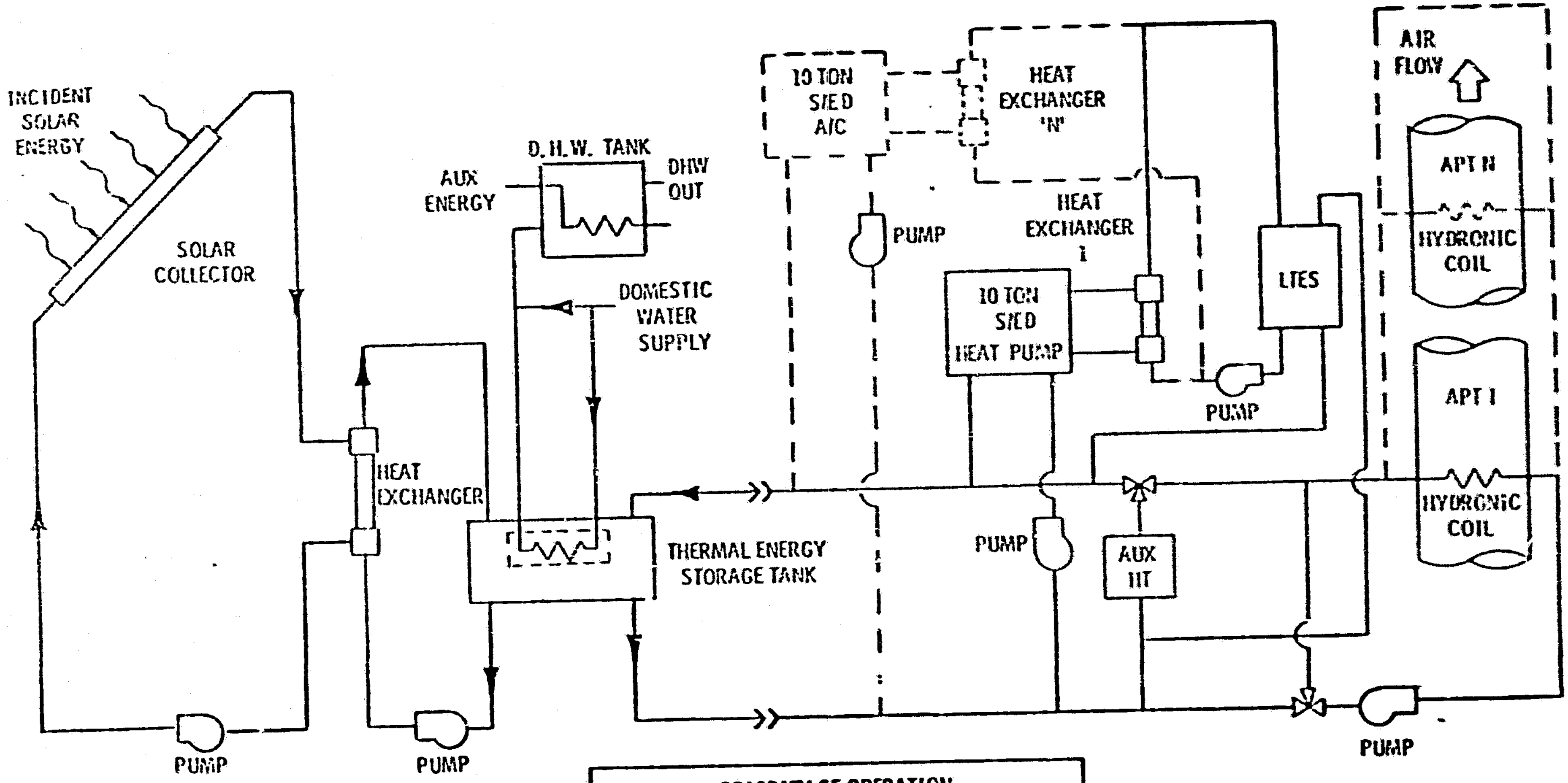
(a) System Configuration Solar Heating, Cooling and Hot Water Single-Family Residential or Small Commercial



PRIORITY OF OPERATION	
HEATING	COOLING
1. DIRECT SOLAR 2. AUXILIARY ENERGY	1. SOLAR DRIVEN 2. ELECTRIC DRIVEN

(b) System Configuration Solar Heating, Cooling and Hot Water Commercial Application (alternate)

Figure 2-5. Solar Cooling System Configuration



PRIORITY OF OPERATION	
HEATING	COOLING
1. DIRECT SOLAR 2. AUXILIARY ENERGY	1. SOLAR DRIVEN 2. ELECTRIC DRIVEN

Figure 2-6. System Configuration Solar Heating, Cooling and Hot Water Multi-family Residential

SECTION 2.3

COMPONENT DESCRIPTION

2.3.1 SOLAR COLLECTOR

The TC-100 solar collector has been developed to a production-ready design for use in a variety of applications.

This collector provides high efficiency over a wide range of energy collection temperatures, insolation levels, and ambient conditions in a design that can be mass-produced at a low cost. It is shown in Figure 2-7. The collector utilizes evacuated glass tubes, a selective coating, a metal reflector, and a circulating heat transfer fluid entirely contained within metal tubing.[1]

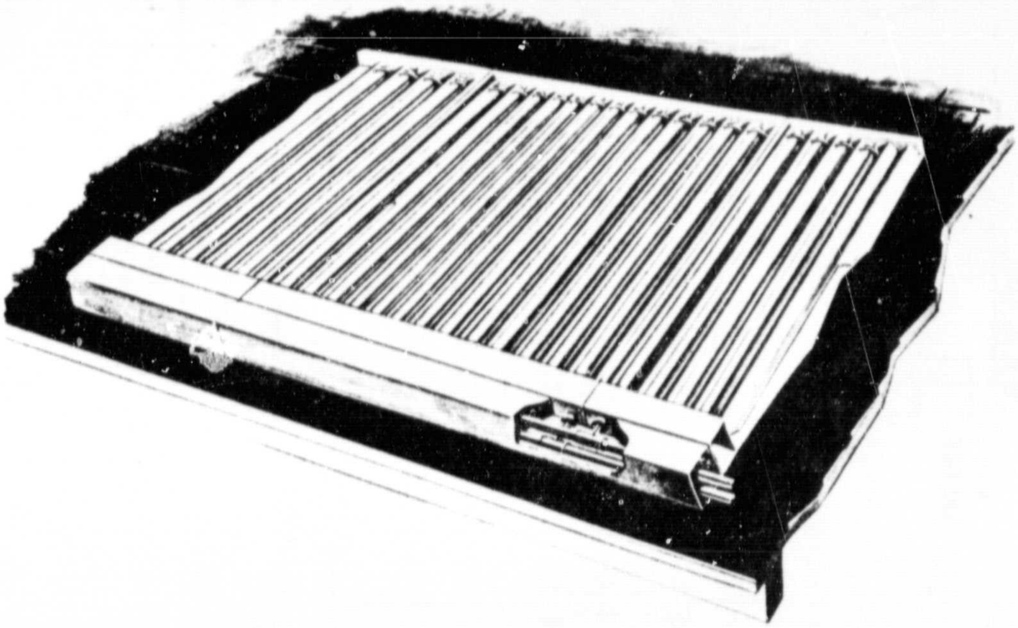


Figure 2-7. TC-100 Vacuum Tube Solar Collector

(1) Information (technical and availability) on the Model TC-100 Vacuum Tube Solar Collector can be obtained from: Manager, Solar Heating and Cooling Marketing, Advanced Energy Programs, General Electric Co., P.O. Bos 13601, Philadelphia, PA. 19101

The performance of the collector is shown in Figures 2-8 and 2-9 and the performance characteristics are listed in Table 2-1. A curve fit of data from DSET, Inc. produced the following efficiency equation

$$\eta = 0.574 - .1313\frac{\Delta T}{I} - .1833\left(\frac{\sigma Ti^4 - \sigma Ta^4}{I}\right)$$

The collector units are approximately 4 foot by 4 foot and include mounting holes at the corners. Mounting brackets and interconnecting fittings are available as accessories. Because its fluid temperature operating range goes to 300°F, the TC-100 solar collector can be effectively used for heating, process heat and cooling applications. Pilot production was initiated in late 1977.

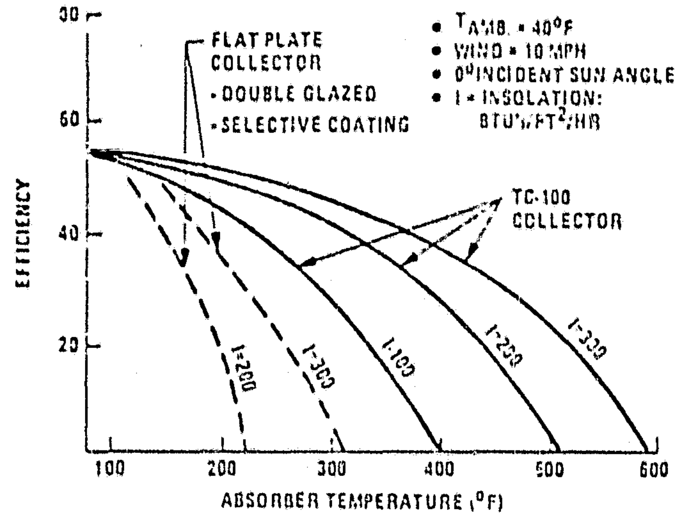


Figure 2-8. Collector Performance

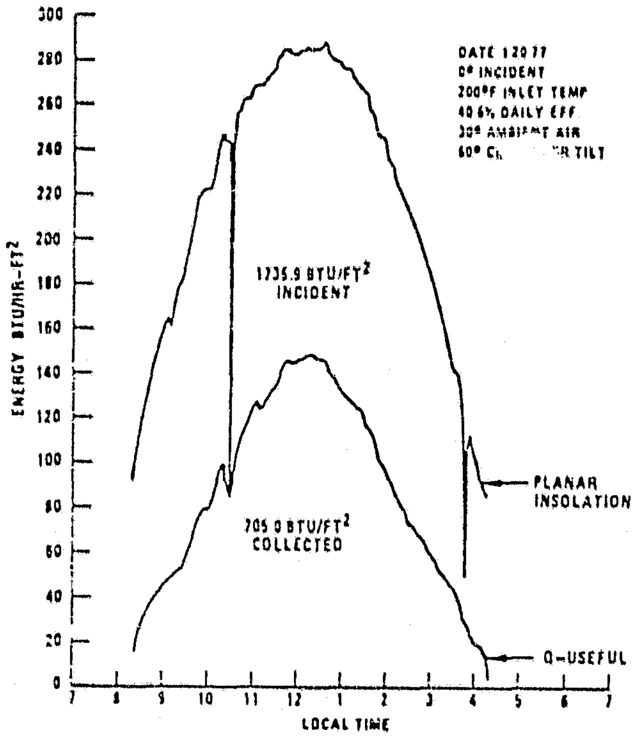


Figure 2-9. Daily Collector Performance Test Results

Table 2-1. Performance Characteristics

- Operational Ambient: -20°F to 140°F
- Operational Fluid Temperature Range: 100°F to 300°F
- ΔP/Collector: Maximum 10 psi @180°F
  Minimum 5 psi
- Maximum Operational Pressure: 125 psig
- Absorber Coating: Selective

The working fluid is a mixture of Prestone II and water.

2.3.2 SOLAR INTEGRATOR

A new component was developed to provide solar energy collection control signals as available equipment did not meet system requirements. The vacuum tube collectors have low thermal mass and very low losses. They will reach relatively high temperatures even with insolation rates so low that useful amounts of energy cannot be collected. Thus, temperature in the collector is not a reliable method for turning on the collector loop. It would turn on because of temperature but flow would lower the temperature because of insufficient insolation and undesirable on-off cycling would occur. These conditions would occur each morning and evening and also could be caused by cloud cover. The sensor provides an average value of insolation and is configured to turn on at the minimum useful level for energy collection based on a time average of the insolation. By analysis the value has been established as 35 Btu/hr-ft^2. The unit, called solar integrator, is set at this nominal value and can be adjusted.for higher values of insolation by the addition of a resistor across the sensor terminals.

The unit also provides a safety lockout in the event of power failure to prevent thermal shocking of the collector fields. The device, shown in Figure 2-10 mounts on the frame of the solar collector.

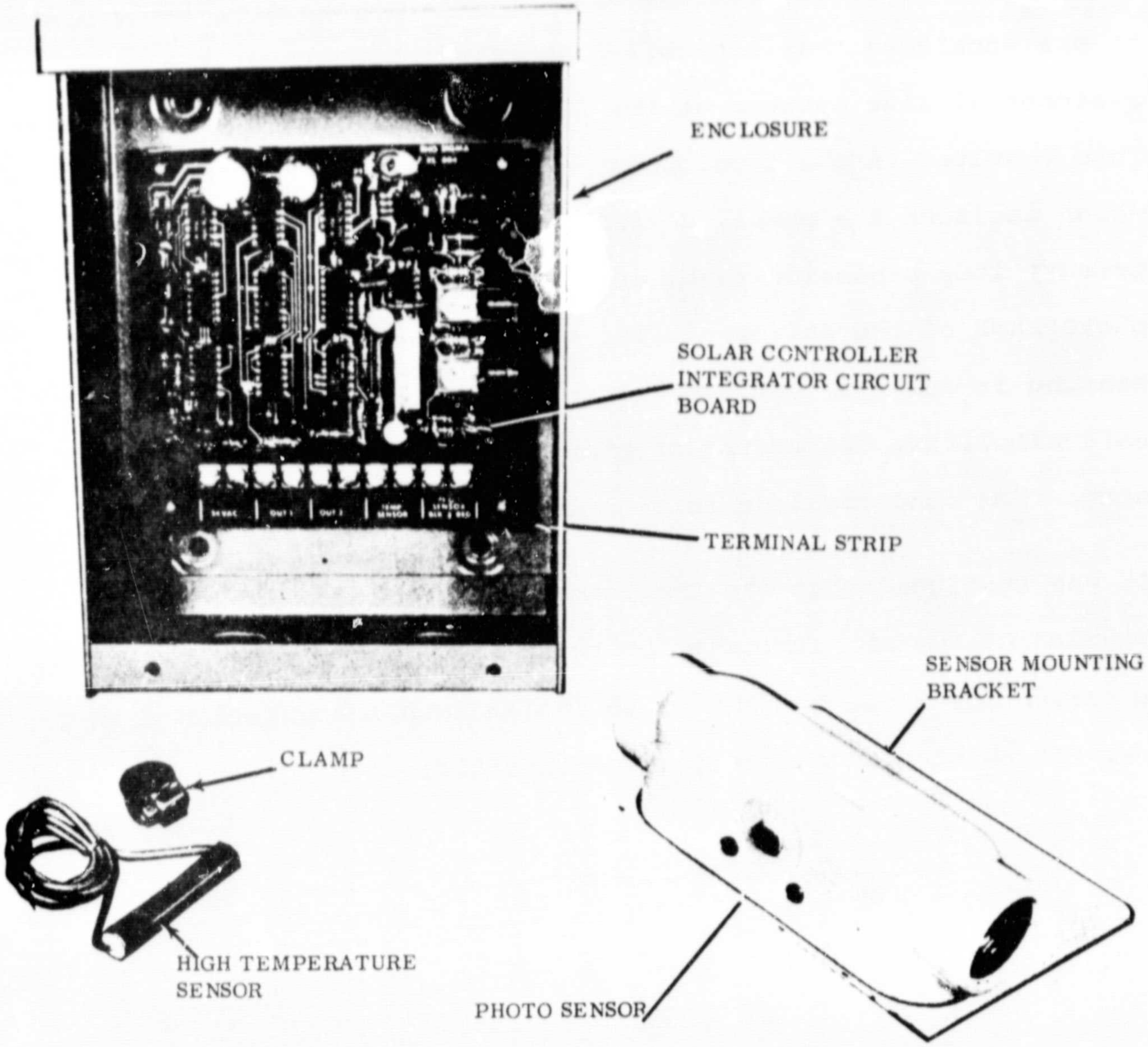


Figure 2-10. Solar Integrator

2.3.3 ENERGY MANAGEMENT MODULE

As a result of investigations of the costs of solar systems, it was concluded that assembling selected elements of the residential size systems at the factory was cost effective. This resulted in the development of an Energy Management Module which included the pumps, control valves, primary heat exchanger, primary loop expansion tank, and electrical controls. A photograph of the unit exhibited at the January, 1978 ASHRE meeting in Atlanta, Georgia, is shown in Figure 2-11. This unit simplifies the amount of piping that has to be done at the site. The functional definition is given in Figure 2-12.

It was concluded that the EMM concept was not attractive for commercial systems at least for the present. The program approach for these systems is to design each installation with respect to component location and installation.

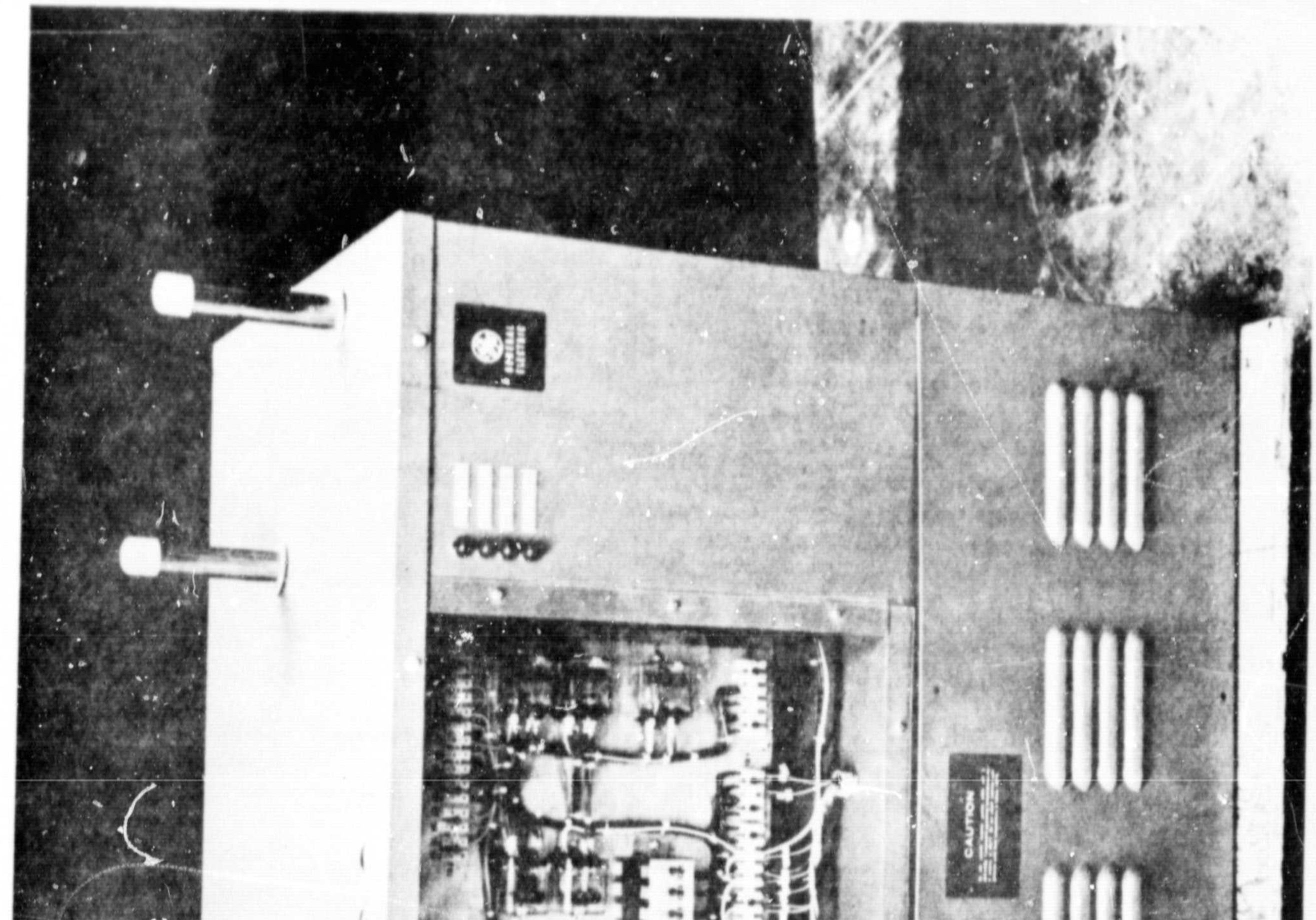


Figure 2-11. Energy Management Module

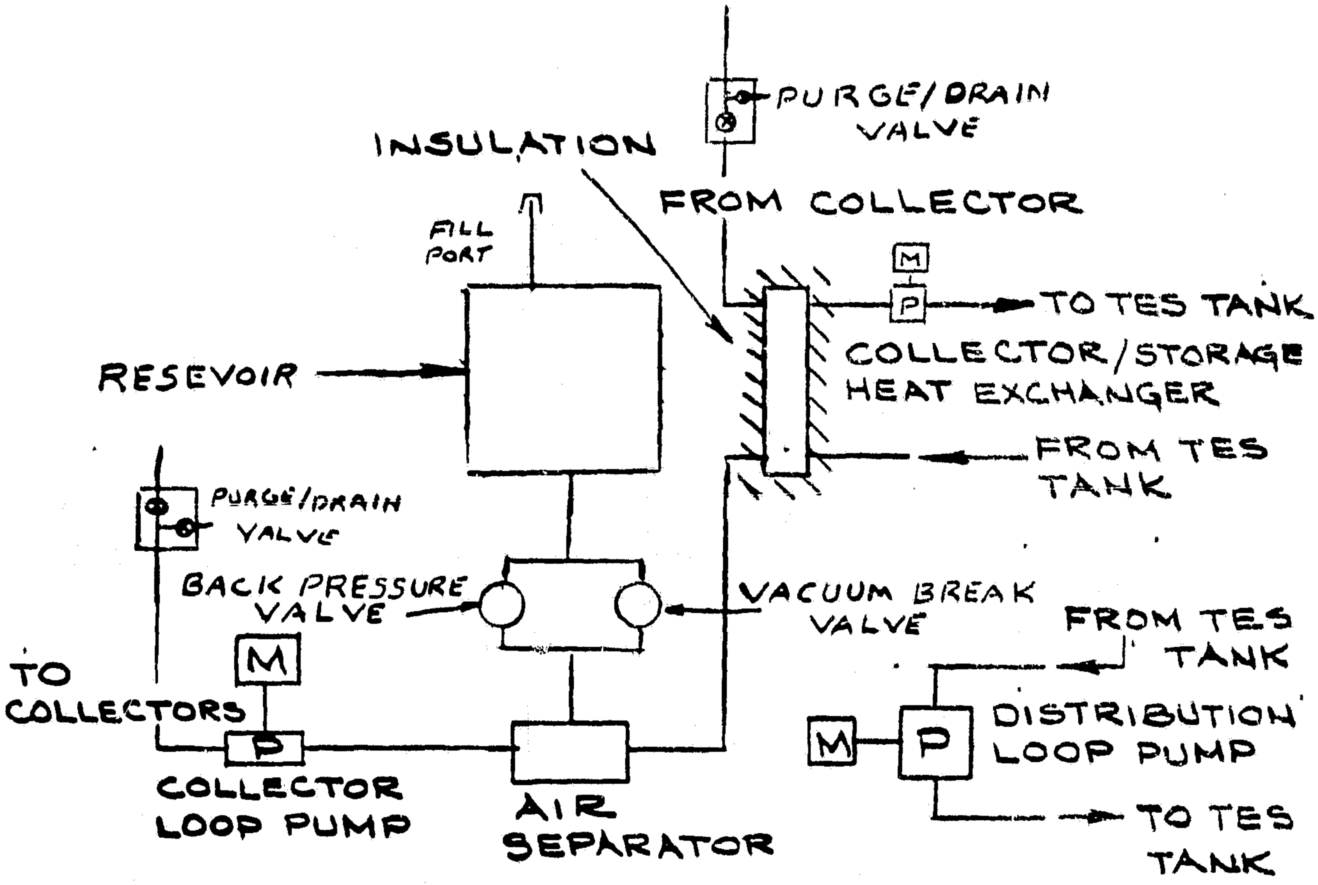


Figure 2-12. EMM Functional Definition

2.3.4 THERMAL ENERGY STORAGE TANKS

Thermal energy storage tank configurations have been developed for both horizontal and vertical installations. The tanks do not rely on thermal stratification for proper operation but do have the porting that takes advantage of stratification when it occurs. The hottest fluid from the heat exchanger enters the tank at the top and at one end (left side). The return to the heat exchanger comes from the bottom at the opposite side of the tank (right side). This promotes full use of the tank. Similarly, hot water sent to the load comes from the top opposite to the heat exchanger inlet (right side) and the return goes to the middle of the tank on the left side. The tanks also have provisions for a DHW coil and ports for control sensors. Typical TES tank configurations are shown in Figure 2-13.

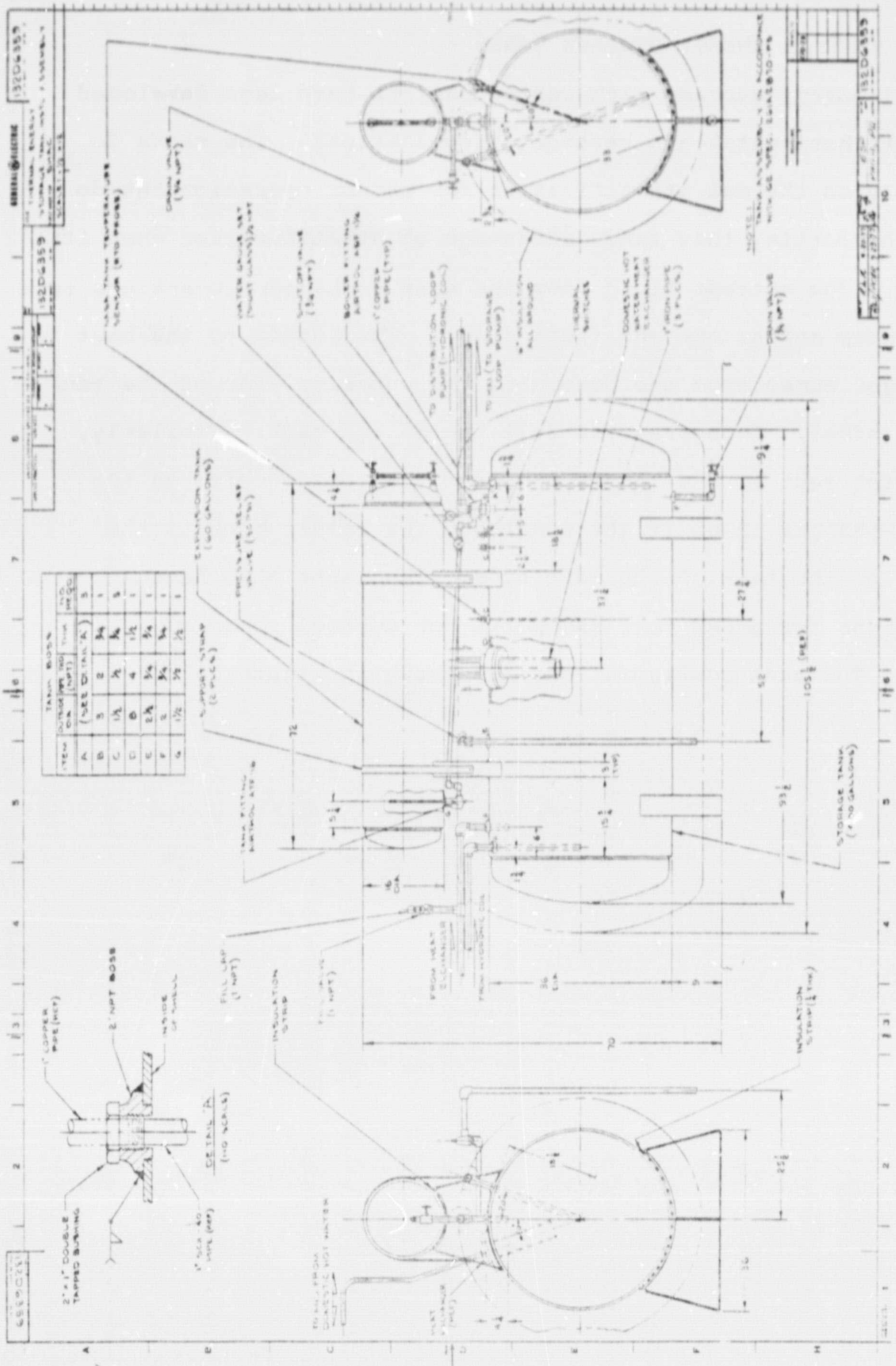


FIGURE 2-13 132D6359

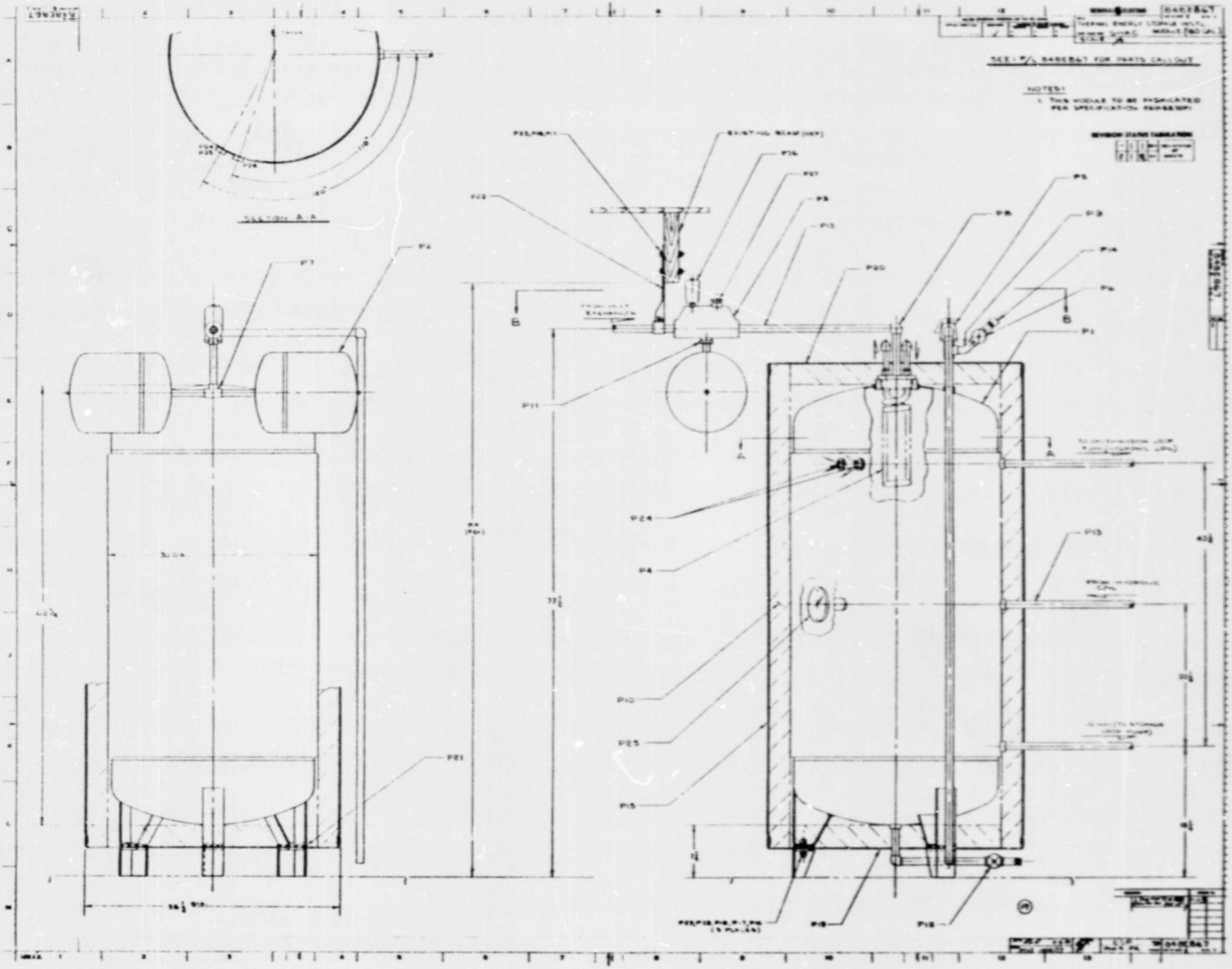


FIGURE 2-13 848E867

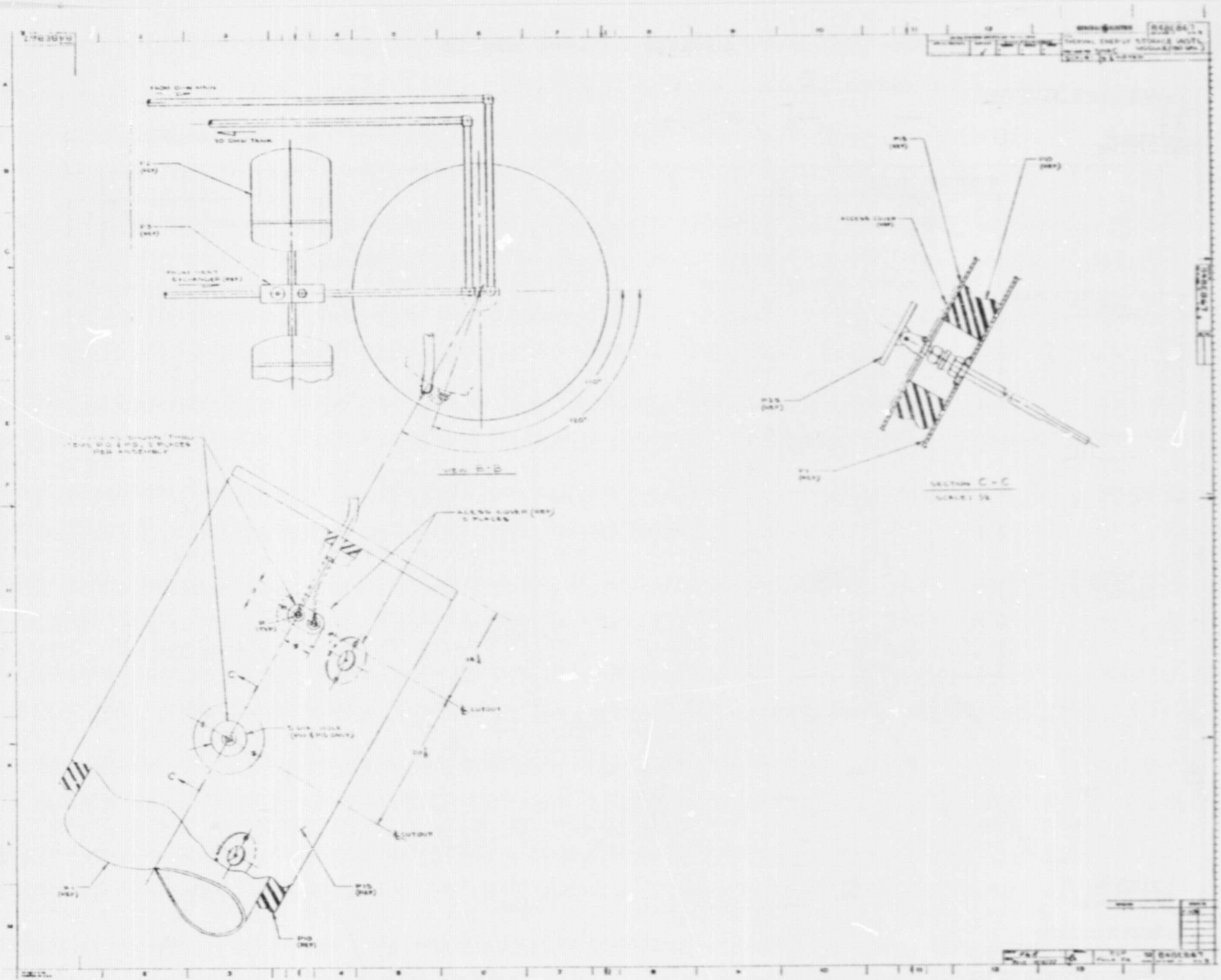


FIGURE 2-13 848E867

2.3.5 RANKINE/ELECTRIC HEAT PUMPS

The Rankine/electric heat pump (REH-30/100)* is a Rankine engine driven vapor compression air conditioning system designed to provide 3 and 10 tons cooling capacity respectively at ARI standard rating conditions. The compressor can be driven by the Rankine engine or by a conventional electric motor. The REH is an air-to-air cooling system with refrigerant R-22 being the air conditioner working fluid. The General Electric Rankine engine is designed to use fluorocarbon FC-88 as the working fluid. A schematic diagram of the REH is shown in Figure 2.3.5-1.

2.3.5.1 Outdoor Unit

The outdoor unit (ODU) is designed for roof top or ground level mounting. For REH 100, the overall size is 81" x 110" x 68" high, and the approximate weight is 8000 pounds. Figure 2.3.5-2 is an external view of the ODU and Figure 2.3.5-3 shows the several service connections for REH-100. Figure 2.3.5-4 is an external view of the REH-30 and Figure 2.3.5-5 shows the several service connections.

As indicated on Figure 2.3.5-1 the Condensing Coils for the REH are integrated into a single fin-tube assembly in a manner to achieve equal air flow and temperature distribution, and to achieve "fin sharing" on the air conditioner side when the Rankine engine is inoperative. "Fin sharing" provides a significant improvement in EER when the air conditioner is operating in the electric mode. The condenser coil assembly is U-shaped (three sided) with the

*REH-30 is a 3-ton unit; REH-100 is a 10-ton unit.

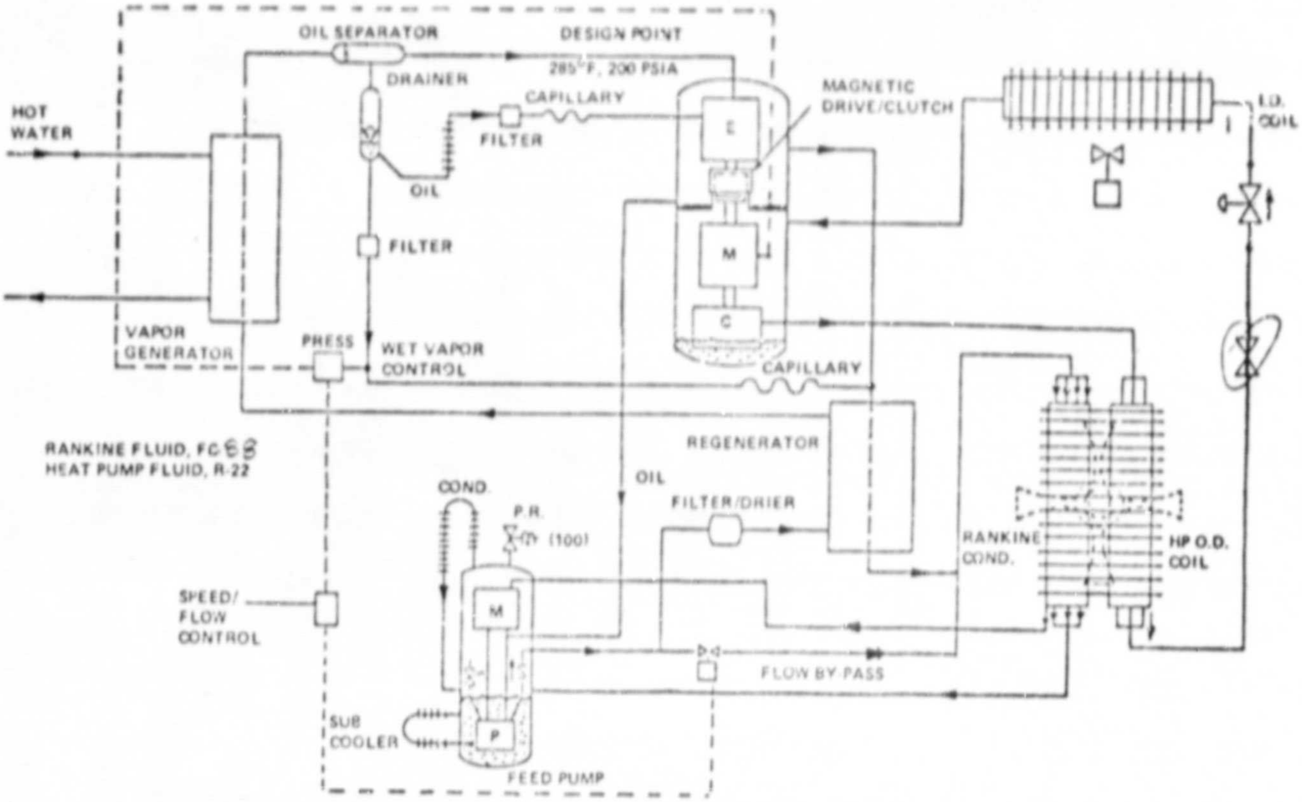


Figure 2.3.5-1. REH Schematic

Figure 2.3.5-2. Outdoor Unit for REH-100 (10 ton)

Rankine engine and air conditioner machinery compartment closing the fourth side of the coil. Cooling air is pulled through the coil by a single two speed fan as shown on Figure 2.3.5-2. High speed fan operation is encountered when the Rankine engine is driving the air conditioner. In electric mode operation, the condenser fan is on low speed.

All Rankine engine components are located in the ODU, and for the REH-100 except for the condenser and fan, all Rankine engine and air conditioner components are inside a sound insulated machinery compartment at one end of the ODU. The Rankine engine, as well as the heat pump, is designed for hermeticity - there are no shaft seals. As shown on Figure 2.3.5-6, the Rankine engine expander is a two-stage multivane expander which is coupled to a multi-piston vapor compressor by a permanent magnet drive and over-running clutch assembly. The expander and compressor run at synchronous speed. The magnetic drive provides hermeticity for the Rankine engine and for the refrigeration system. The over-running clutch self-engages when the Rankine expander starts to rotate.

As indicated on Figure 2.3.5-1 the Rankine engine feed pump is driven by an electric motor. The pump and motor are incorporated within a steel pressure vessel which also serves as a liquid receiver for the Rankine engine working fluid. The pump has two stages - the first stage being a centrifugal impeller for superior

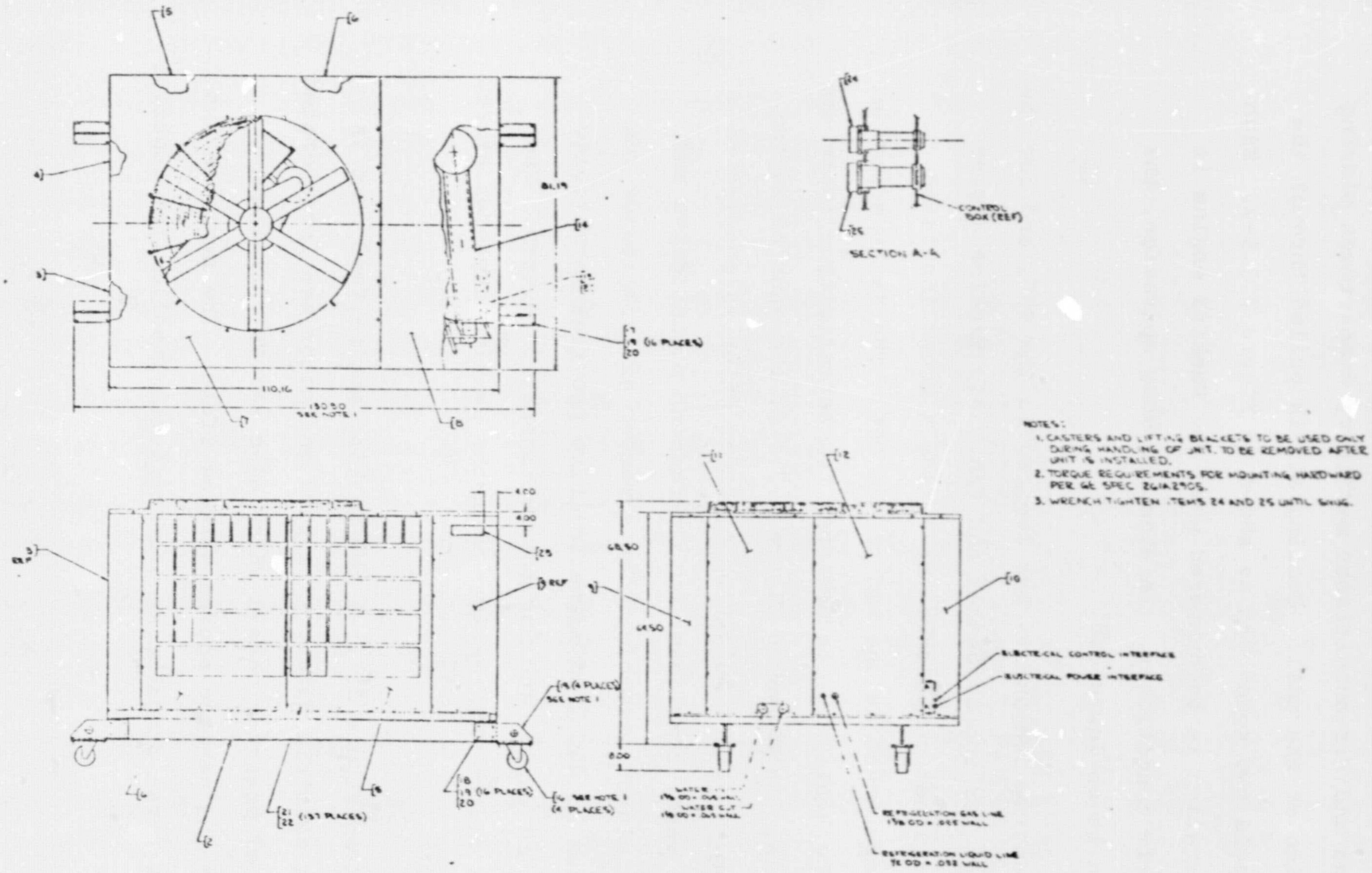


Figure 2.3.5-3. Outdoor Unit Top Level Assembly for REH-100

pumping performance at near zero net positive suction head. The second stage is an internal gear pump element. A cross sectional view of the Rankine engine feed pump is shown in Figure 2.3.5-7.

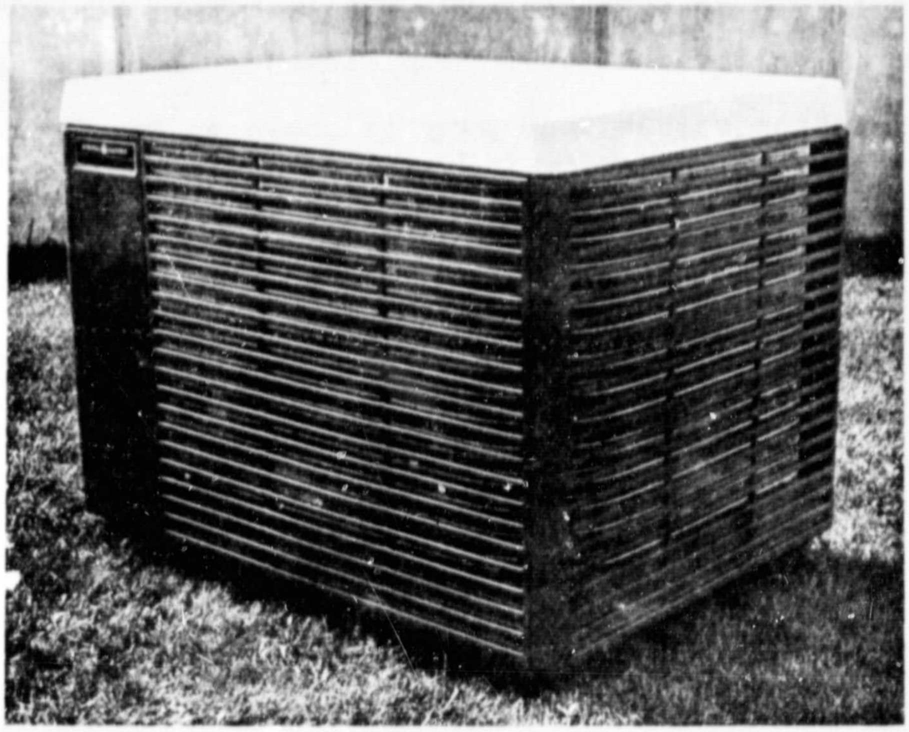


Figure 2.3.5-4. Outdoor Unit for REH-30 (3 ton)

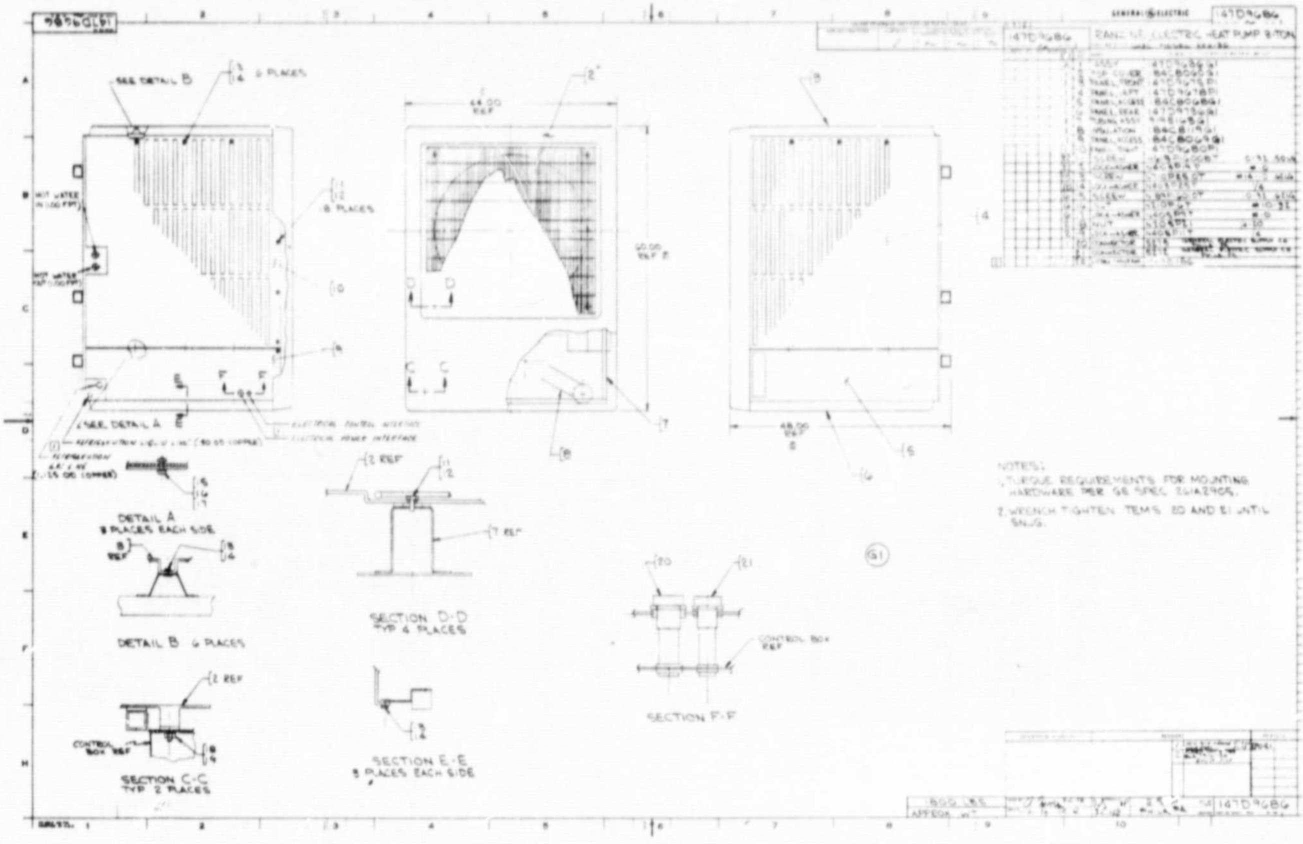


Figure 2.3.5-5. Outdoor Unit Top Level Assembly for REH-30

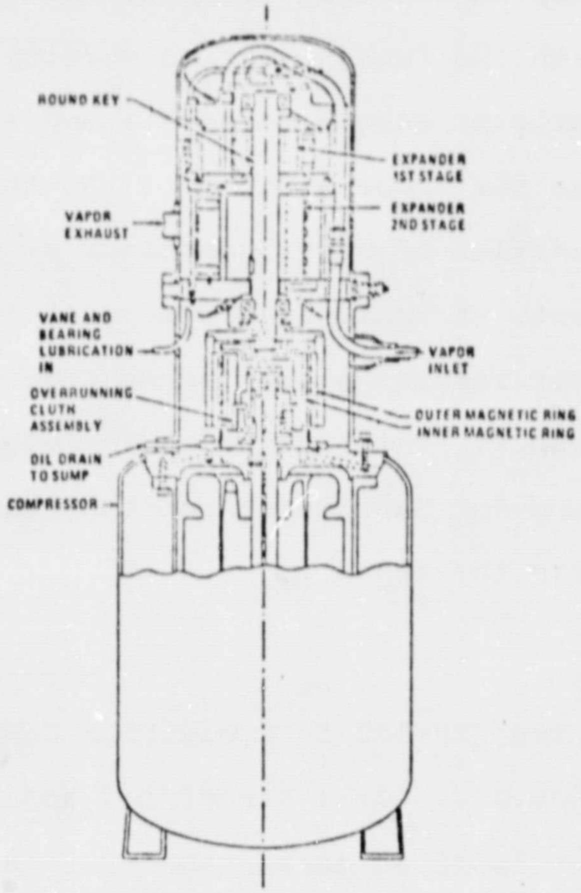


Figure 2.3.5-6. Rankine Expander/Compressor Assembly

A single unit heat exchanger is used for the Rankine engine fluid preheater, vaporizer and superheater. This heat exchanger is a single pass, counter-flow tube and shell unit with the heat transport fluid (water) on the shell side. The heat exchanger is compact and insulated for low heat loss. The Rankine engine regenerator is also a tube and shell heat exchanger, single pass, counter flow, with straight tubes. Liquid is on the shell side for lower liquid inventory and lower heat loss.

Expander and feed pump lubrication is provided by a lubricant which is miscible with the Rankine engine working fluid. The concentration of lubricant ranges between 3 and 4% as the combined fluids enter the vapor generator. At this concentration no appreciable degradation of vapor generator or condenser performance is evident. A small liquid separator at the outlet of the vapor generator removes a high percentage of the liquid lubricant from the vapor. This lubricant is metered to the expander and feed pump for lubrication of bearings and for viscous sealing within the expander.

2.3.5.2 Air Handler

The air handler for the REH-100 is a modified General Electric unit, Model No. BRB006-016. Coil circuiting and air flowrate have been altered for improved performance for the solar powered air conditioner. In effect, condensing temperature has been set at 50°F for the solar air conditioner as compared to approximately 45 F for conventional straight electric units. Overall dimensions for the air handler are 81-1/2" x 63" x 37-1/2" high. Installed weight is 740 pounds. Typical installations for the REH-100 air handler are shown on Figure 2.3.5-8. An overall view of the REH-3 air handler is shown in Figure 2.3.5-9.

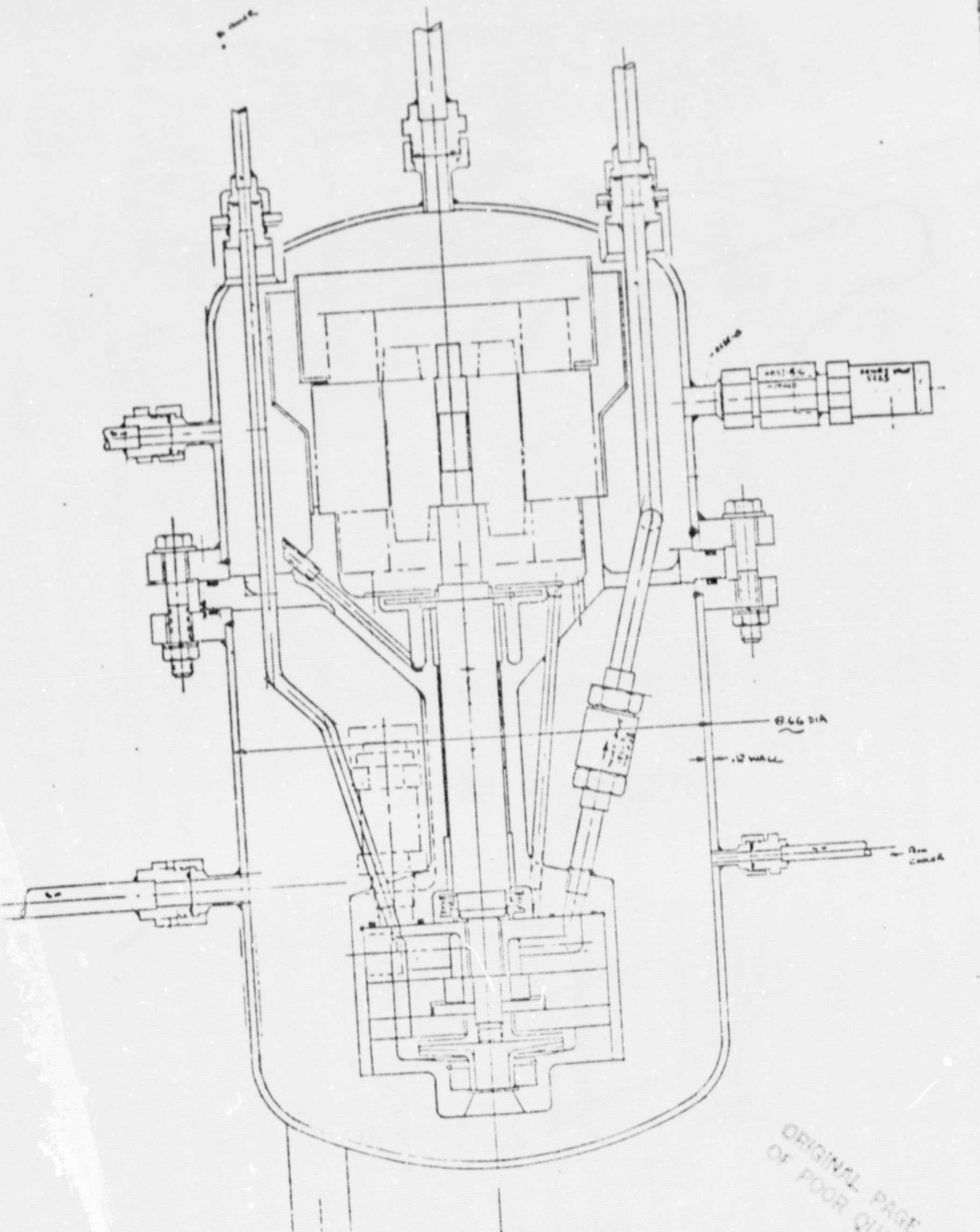
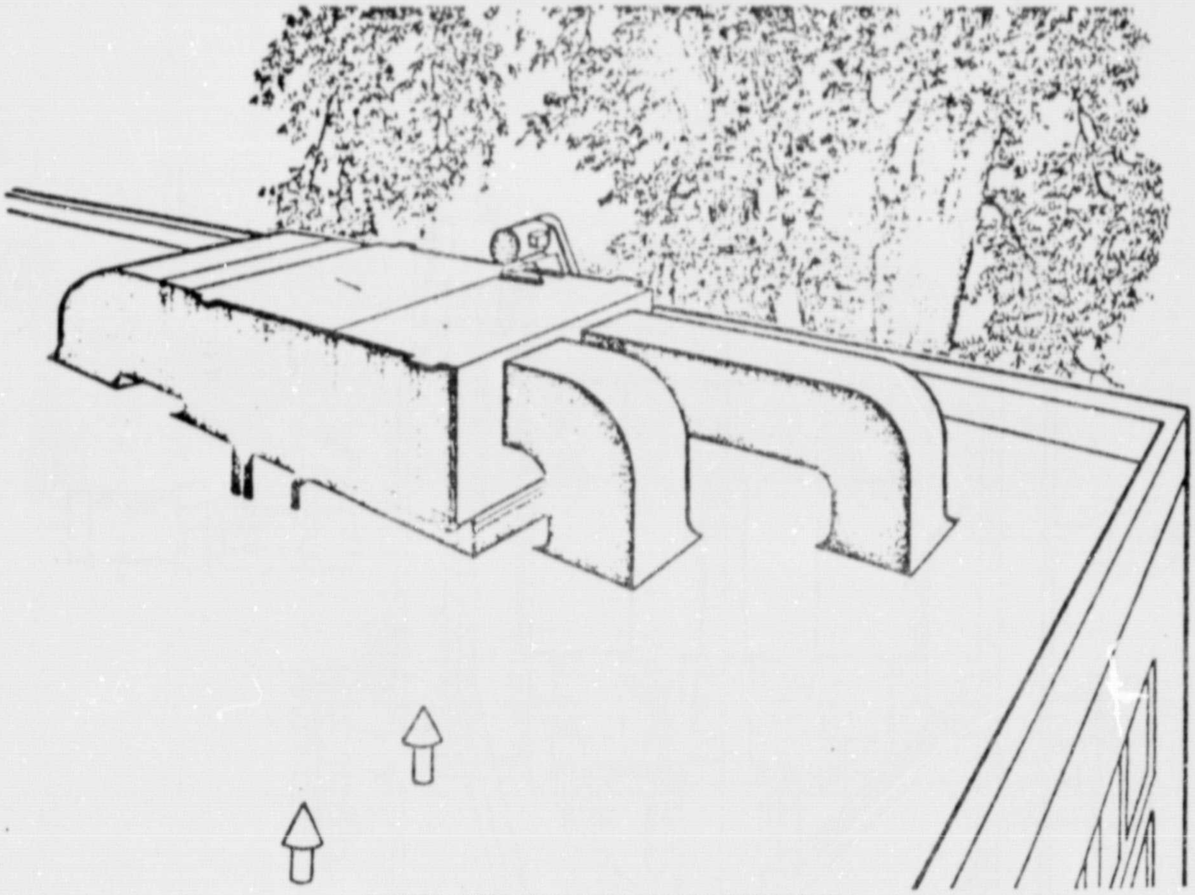
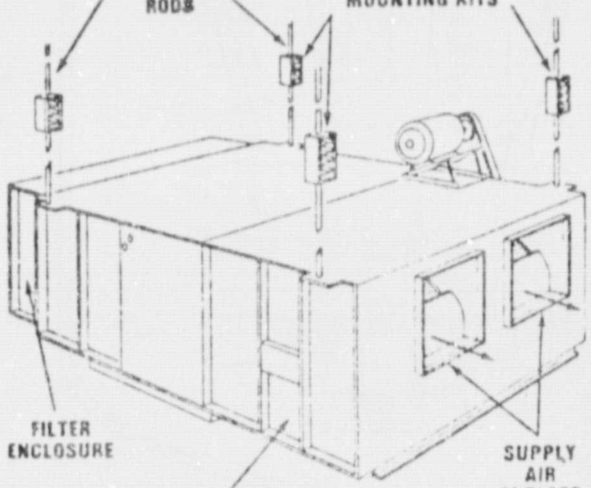


Figure 2.3.5-7. Hermetic Feed Pump



TYPICAL "WEATHERPROOFED" ROOF INSTALLATION



HORIZONTAL UNIT

Figure 2.3.5-8. Air Handler Installation

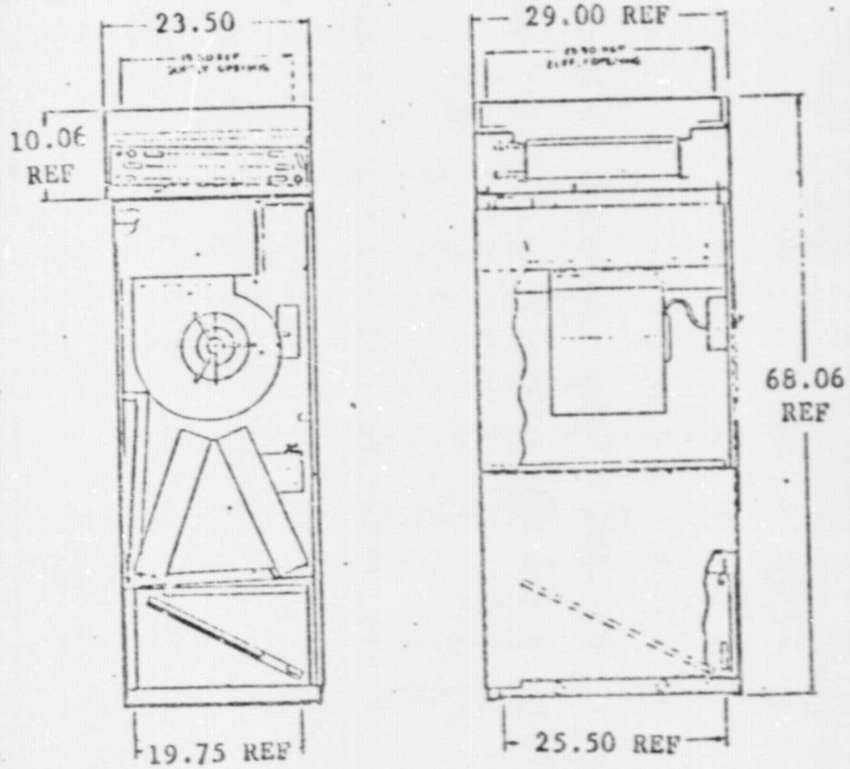


Figure 2.3.5-9. REH-30 Air Handler

2.3.6 CONTROLS

The control logic for the solar systems has been developed to maximize the use of solar energy. The solar collection and storage loops operate from the solar integrator (Section 2.3.2). Pumps P1 and P2 get turned on when the average insolation exceeds a threshold value and get turned off when the insolation goes below that value. A high temperature switch is used to protect the collector and the fluid.

In the early systems, using a heat dump, if the fluid discharge temperature exceeds a set point (usually 280^{0}F), P2 will shut off and the collection loop goes into a heat dump mode. A high temperature switch (CT4) in the TES tank will accomplish the same thing.

In the later systems, if the temperature of the fluid exceeds the set point, pumps P1 and P2 shut down. The collector fluid will boil inside the collector and will leave by virtue of the high pressure. This expelled fluid will be collected in the expansion tank, TKX-1.

The heating system controls are contained in the EMM for residential applications but incorporated in a control box for commercial sites. A control logic table for heating only residential system is shown in Table 2-1.

TABLE 2-I

CONTROLS COLLECTION - MODES

SOLAR INTEGRATOR						
1st	2nd	T1, T4	P1	P2	V4	V5
OFF	OFF	X	OFF	OFF	OFF	OFF
ON	OFF	X	ON	OFF	OFF	OFF
ON	ON	WITHIN LIMITS	ON	ON	ON	ON
ON	ON	OUT OF LIMITS	ON	OFF	OFF	ON

CONTROLS - UTILIZATION MODES

THERMOSTAT						
1st	2nd	FAN MANUAL	T3(1)	DISTRIBUTION PUMP, P3	AUXILIARY FURNACE	FAN
OFF	OFF	AUTO	X	OFF	OFF	OFF
ON	OFF	X	$>$ Tmin	ON	OFF	ON
ON	OFF	AUTO	$<$ Tmin	OFF	ON	ON
ON	ON	AUTO	$>$ Tmin	OFF	ON	ON
OFF	OFF	ON	X	OFF	OFF	ON

(1) Tmin = Lower Limit of Solar Heating

Heating and cooling systems use electric heat pump heating as a back up mode. GE's solar heating control logic is based on maximizing the use of solar energy. A three stage thermostat developed by GE is used in this application. A control logic table for the solar heating and cooling systems is shown in Tables 2-2 and 2-3. A differential controller is added to control the storage loop pump to prevent loss of energy when operating at the high temperatures required for air conditioning.

TABLE 2-2

SYSTEM MODE TABLE

SOLAR COLLECTION SYSTEM

SOLAR INTEGRATOR	TEMPERATURE SENSOR	SYSTEM MODE	COLLECTOR TO TES ΔT (NOTE 1)	P1	P2
OFF	X	X	X	OFF	OFF
ON	(NORMAL) CLOSED	COOLING	$> 20^{\circ}$	ON	ON
ON	(NORMAL) CLOSED	COOLING	$< 3^{\circ}$	ON	OFF
ON	(NORMAL) CLOSED	HEATING	X	ON	ON
X	OVER TEMPERATURE OPEN	X	X	OFF	OFF

NOTE 1:

THE DIFFERENTIAL THERMOSTAT CIRCUIT HAS HYSTERISIS. IT REQUIRES MORE THAT 20° TO TURN ON AND LESS THAT 3° TO TURN OFF.

TABLE 2-3

SYSTEM MODE TABLE: HEATING AND COOLING SYSTEM

MODE SWITCH (NOTE 5)	STAGE	CT3	CT5	CT6	AMBIENT TEMP.	OUT DOOR FAN	IN DOOR FAN	SWITCH OVER VALVE	P3	V4	ELECT. COMP.	RES. HEAT	RANKINE PUMP	
OFF	X	X	X	X	X	OFF	OFF	ON (C)	OFF	REH 100	OFF	OFF		
H	H1	< 95°F		> 40°F		LO	HI	OFF(H)	OFF		ON	OFF		NOTE 1
		> 105°F						OFF(H)	ON	HYDRO COIL	OFF	OFF		
		> 105°F			> 40°F			OFF(H)	ON	HYDRO COIL	ON	OFF		NOTE 1
		≥ 105°F			≤ 40°F			OFF(H)	ON	HYDRO COIL	OFF	ON		
H	H2	< 95°F			> 40°F			OFF(H)	OFF	HYDRO COIL	ON	OFF		NOTE 1
		< 95°F			< 40°F			OFF(H)	OFF	HYDRO COIL	OFF	ON		
H	H3	> 105°F			> 40°F			OFF(H)	ON	HYDRO COIL	ON	ON		NOTE 1
	H3	> 105°F			< 40°F			OFF(H)	ON	HYDRO COIL	OFF	ON		
	H3	< 95°F			> 40°F			OFF(H)	OFF	HYDRO COIL	ON	ON		NOTE 1
	H3	≤ 95°F			< 40°F			OFF(H)	OFF	HYDRO COIL	OFF	ON		
H	0,1,2 or 3	X	X	< 40°F		LO	HI	OFF	ON	REH 100	X	X		
C	OFF	X	X	X	X	OFF	OFF	ON (C)	OFF	REH 100	OFF	OFF		
C	C1 NOT CONNECTED													
C	C2		< 240°F	> 40°F		LO	LO	ON (C)	OFF	REH 100	ON	OFF		NOTE 4
C	C2		> 240°F			HI	HI	ON (C)	ON	REH 100	OFF	OFF	ON	NOTE 2
C	C3		> 240°F			LO	LO	ON (C)	OFF	REH 100	ON	OFF	OFF	NOTE 3

NOTE:
1. REH 100 UNIT MAY DEFROST WHICH WILL SET SWITCH OVER VALVE TO COOLING MODE AND STOP OUT DOOR FAN.
2. CT5's OUTPUT ABOVE 240°F WILL LATCH UNTIL C2 IS SATISFIED.
3. THIRD COOLING STAGE WILL CAUSE UNIT TO SWITCH TO ELECTRIC MODE AFTER 16 MINUTES OF RANKINE MODE OPERATION.
4. UNIT WILL SWITCH TO ELECTRIC MODE IF UNABLE TO MAINTAIN MININUM EXPANDER SPEED.
5. AUTO POSITION IS NOT OPERATIVE.
6. ONLY RESISTANCE HEAT IS SELECTED WITH EMERGENCY HEAT SWITCH.

Section 3

OPERATIONAL TEST SITES

3.1 SITES EVALUATED

Prototype systems have/will be installed in Operational Test Sites listed in Table 3-1. The two heating and cooling commercial systems have not been installed at the time this report is written.

Table 3-1. Operational Test Site Summary

System Type	Location	Application	Nominal Collector Area (ft^2)
Heating-Single Family	Normal, Ill.	Farm House	320
Heating-Single Family	Ft. Meade, Md.	Residence	384
Heating-Commercial	Milwaukee	Community Ctr.	1392
Heating-Commercial	Spokane, Wash.	YWCA	4800
Heating and Cooling-Single Family	Dallas, Texas	Residence	336
Heating and Cooling-Commercial	Muscle Shoals, Ala.	Records Ctr.	1536
Heating and Cooling-Commercial	Murphy, NC	School for Handicapped	1536

Originally 12 sites were planned but the four multi-family sites and one heating and cooling single family site were cancelled prior to ordering hardware for any of them. The sites inspected by GE from which those seven resulted are listed in Table 3-2.

Table 3-2. Sites Inspected by GE

Bldg. Type	General City	Site
HSF	Baltimore	7502 Young St., Ft. Meade, MD
HCOM	Muscle Shoals	TVA Office at Muscle Shoals, AL
HMF	Nashville	Airman's Quarters, AEDC, Tullahoma, TN
HSF	Peoria	Chanute Air Force Base
HSF	Peoria	MHA, Champaign, Illinois
HSF	Peoria	ISU House, Normal, Illinois
HMF	Schenectady	MHA, Schenectady, NY
HMF	Schenectady	VA Hospital Staff Housing, Albany, NY
HMF	Schenectady	Ely Park Housing, Binghamton, NY
HCMF	Chicago	Ft. Sheridan, ILL
HCMF	Chicago	Great Lakes Naval Training Center
HCOM	Madison	Hill Farm State Office Bldg., Madison, WI
HCOM	Milwaukee	Washington Park Senior Citizens Center
HCOM	Milwaukee	Washington Park Community Center
HCOM	Milwaukee	Dr. Martin Luther King Community Center
HCOM	Spokane	YWCA
HCOM	Spokane	East Washington State College
HCOM	Spokane	Community College
HCCOM	Los Angeles	West L.A. Municipal Building
HCCOM	Los Angeles	Department of Water & Power #1
HCCOM	Los Angeles	Department of Water & Power #2
HCCOM	Los Angeles	Peck Park Recreation Building
HCCOM	Los Angeles	Police Credit Union
HCSF	Dallas	President's Home, Univ. of Texas, Dallas
HCSF	Dallas	President's Home, N. Texas State, Denton, TX
HCSF	Dallas	Grad Student Housing at SMU
HCSF	Philadelphia	Visitor's Center, Valley Forge National Park
HCSF	Philadelphia	Rental House, Valley Forge National Park
HCSF	Philadelphia	Ampitheatre, Valley Forge National Park
HCSF	Philadelphia	Storage Barn, Valley Forge National Park

3.2 TEST SITE DESCRIPTIONS

3.2.1 Ft. Meade, MD Residence

The system installed at Ft. Meade, Md. is the originally designed solar heating system. The site is an atypical retrofit application since the equipment room is very small, the roof has a very shallow pitch, and the thermal storage system is located outdoors in a shed attached to the house. The collectors are mounted at a 40° tilt angle using a support structure attached to the roof. The EMM just fits in the equipment room which also contains an electric hot water heater and a gas fired furnace. Piping between the EMM and the TES tank is run outdoors and is insulated. A system schematic is shown in Figure 3-1 and system size is contained in Table 3-3

3.2.2 Normal, IL Residence

The system installed in the Normal, Illinois, site is a conventiona retrofit application in a residence with fuel oil fired, hot air system and electric heated hot water. The TES tank, EMM and data system are installed in the basement and the collectors are on a dormer added to the frame structure.

The original system installed was the same as that shown in Figure 3-1. Data from the system over the summer of 1978 has provided evidence that boiling was occuring in the collectors at 240°F. The collector loop was modified in December, 1978 to

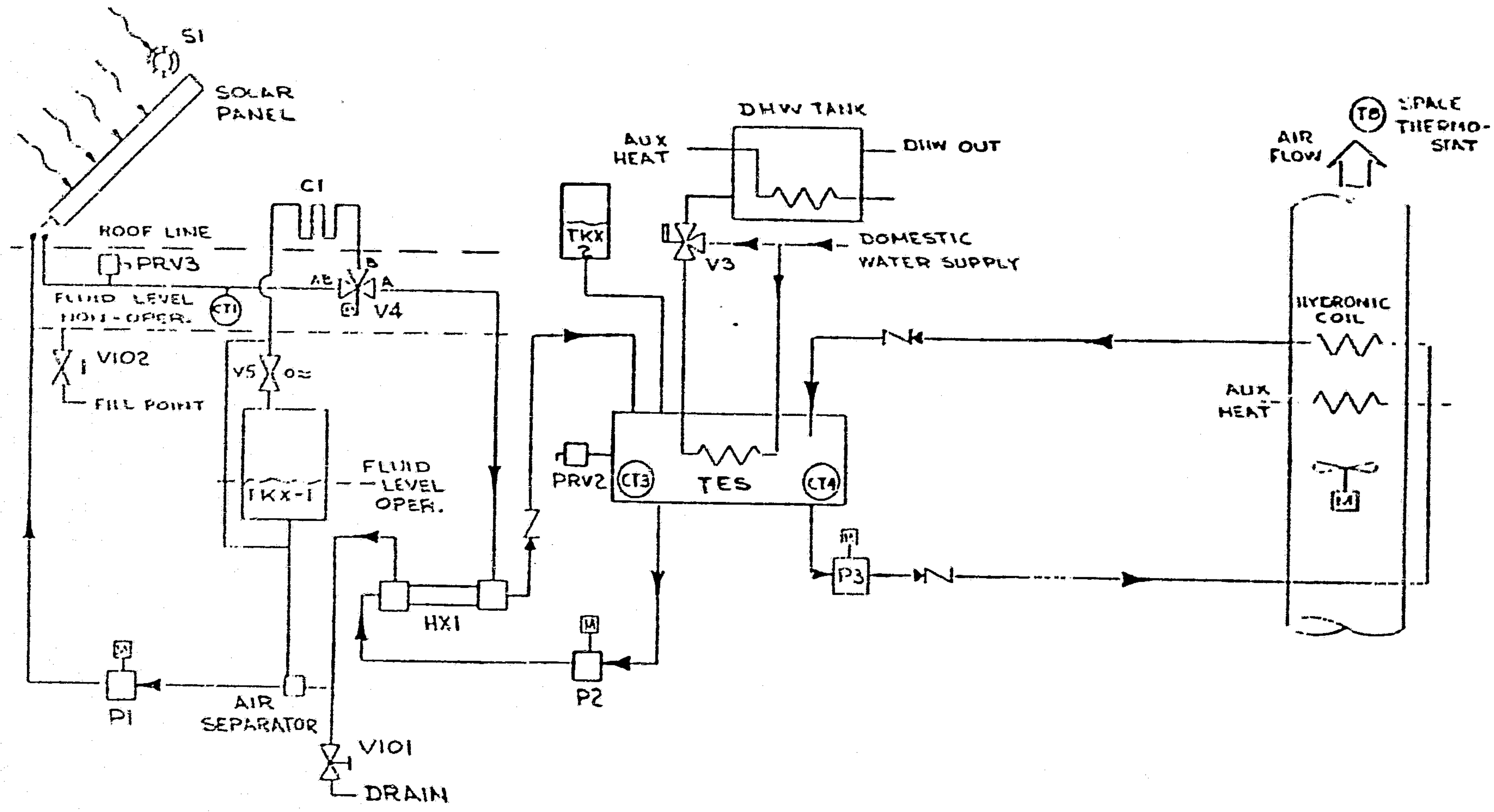


Figure 3-1. Schematic Diagram for Ft. Meade, Md.

Table 3-3

System Sizing Summary

	Ft. Meade	Normal	Milwaukee	Spokane	Dallas	TVA
No. of Collectors	24	20	87	300	21	96
Net Area, ft^2	356	297	1290	4449	311	1424
Tilt Angle	49°	40°	58°	58°	22°	25°
Azimuth Angle (Compass, $\pm 5^{\circ}$)	0	0	0	0	0	0
TES tank size, gal.	380	400	2500	6750	400	1710
Pump Power						
Collector, HP	2 @ 1/12	2 @ 1/12	1	2	2 @ 1/12	3/4
Storage, HP	1/12	1/12	1/2	1 1/2	1/12	1/2

the Loop II design developed for the cooling applications. The revised system schematic is presented in Figure 3-2. The major differences are:

1) elimination of the heat dump system based on collector fin testing at stagnation conditions.

2) addition of valves V1, a back pressure valve used for system pressurization and V2, a vacuum break valve to allow fluid to pass from TKX-1 back to the loop.

The revised loop has been operating satisfactorily since the changeover.

The control system has been modified once when the loop was changed and another time when an air conditioner was added to the system in the summer of 1979.

System sizing is summarized in Table 3-3.

3.2.3 Milwaukee, WI Community Center

The system installed in the Milwaukee, Wisconsin site is a unique retrofit application by virtue of the building being originally designed to include solar collectors on the roof. All other aspects of the application were typical for a commercial retrofit application. The TES tank, pumps, heat exchangers and controls are located in the basement. The expansion tank was located in a first floor closet along with the heat dump tank. The gas fired furnace was non conventional in that it operates with a modulating gas valve and incorporates a 4 zone mixing box for the hot and cold deck air flows. The control system had to be designed expressly for this application.

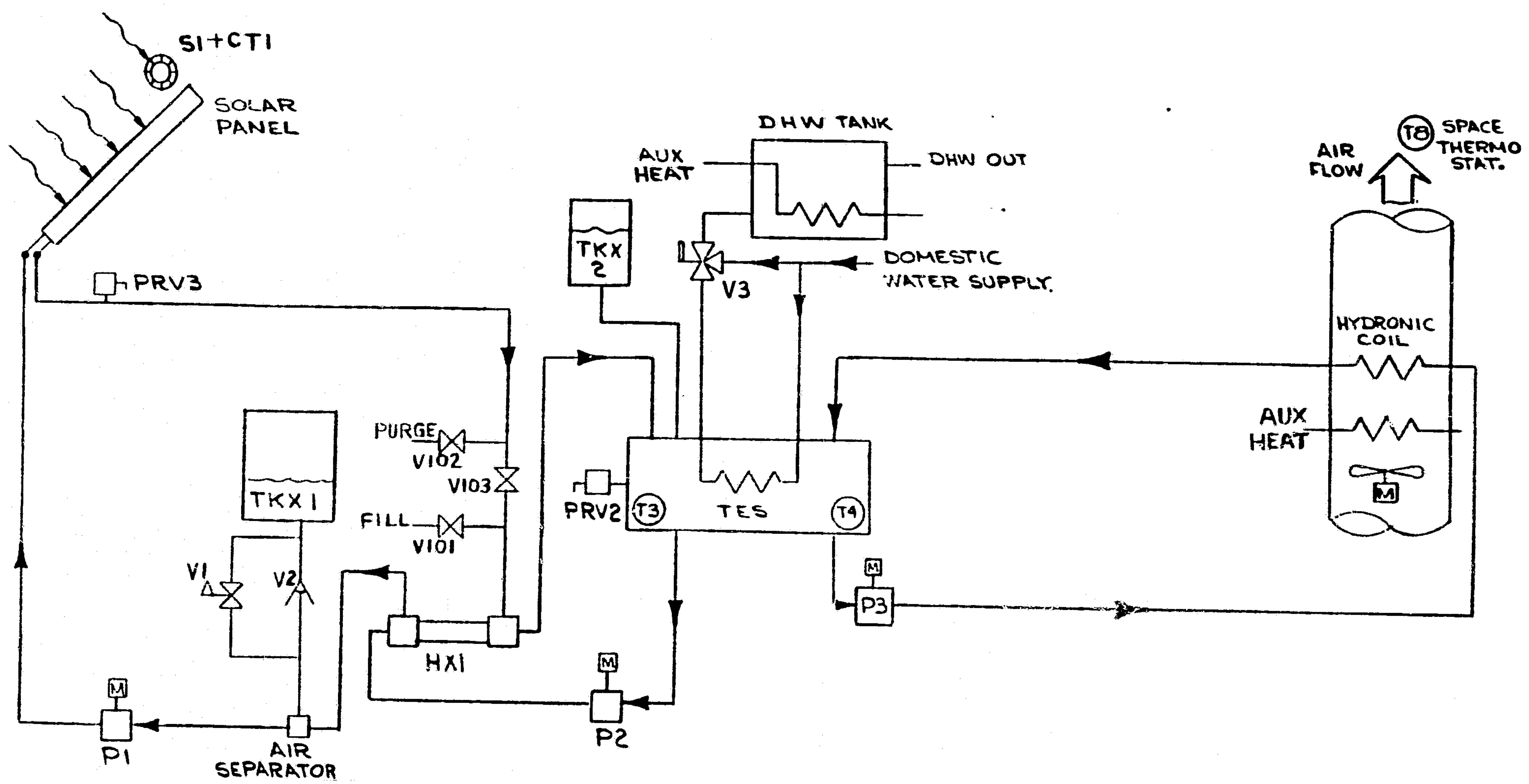


Figure 3-2. Schematic Diagram for the Normal, Il. Operational Test Site System

The system schematic is shown in Figure 3-3. It incorporates the loop I collector loop design which uses a heat dump for over temperature protection. The heating system coil is installed in the hot deck of the air handler downstream of the gas burner. The thermal storage tank is oversized since it was procured for a previous site that had to be abandoned. The system size summary is contained in Table 3-3.

3.2.4 Spokane, Wa. YWCA

The system installed in the YWCA site is another unique retrofit application. The collector array is supported by a structure system added to the roof of the swimming pool wing. In this respect the site is a typical retrofit for flat roofed buildings. However, the thermal storage tank was sized for the intermittent swimming pool heating for therapeutic use during the summer and is well oversized for a typical solar heating system. Also, because of the size of the air distribution system, the existing heating coils were used and solar heating requires the TES tank temperature to be at the same temperature as the boiler, both being controlled by outdoor reset thermostats.

A schematic of the system is shown in Figure 3-4 and system sizing is summarized in Table 3-3. This system was the loop I design which incorporates the heat dump for overtemperature protection. In the summer of 1979, the heat dump became ineffective due to clogged inlet water lines to its tank. As a result the system has been rewired to use sensor CT1 to shut the system off if

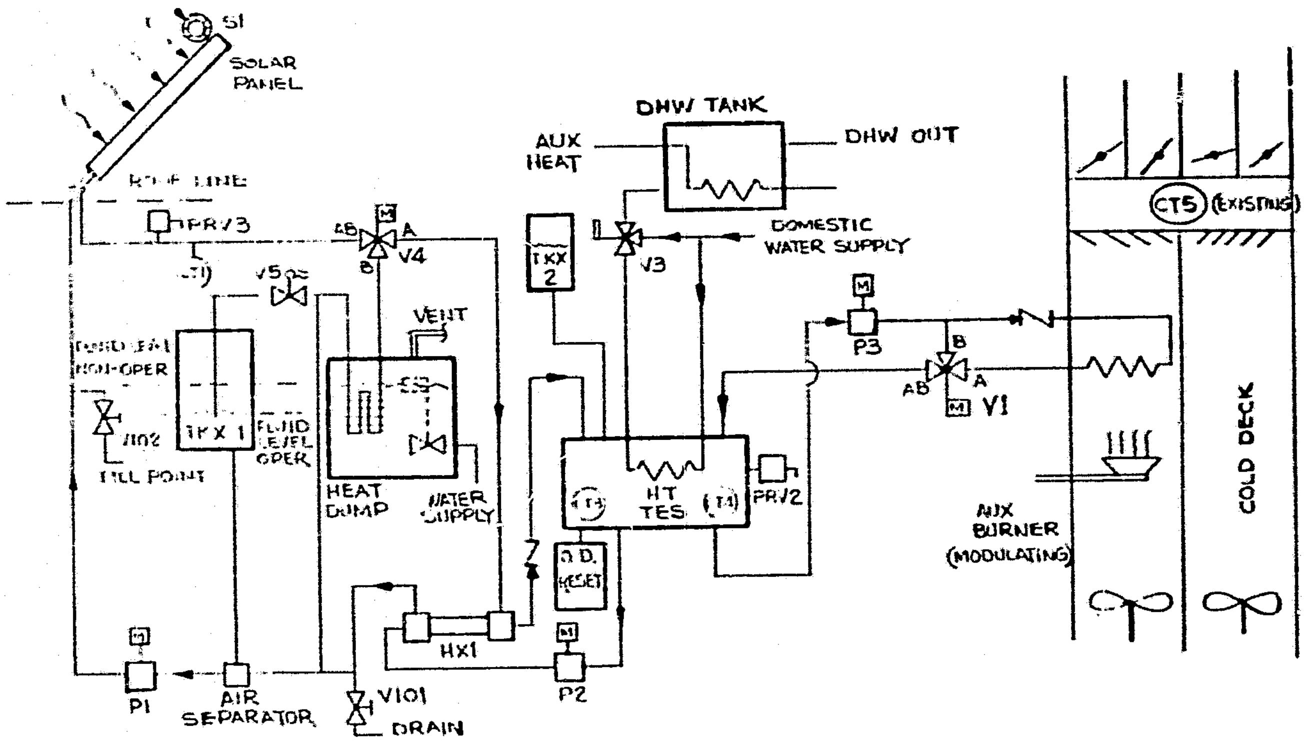


Figure 3-3. Schematic Diagram for the Milwaukee, Wisconsin Solar Heating System

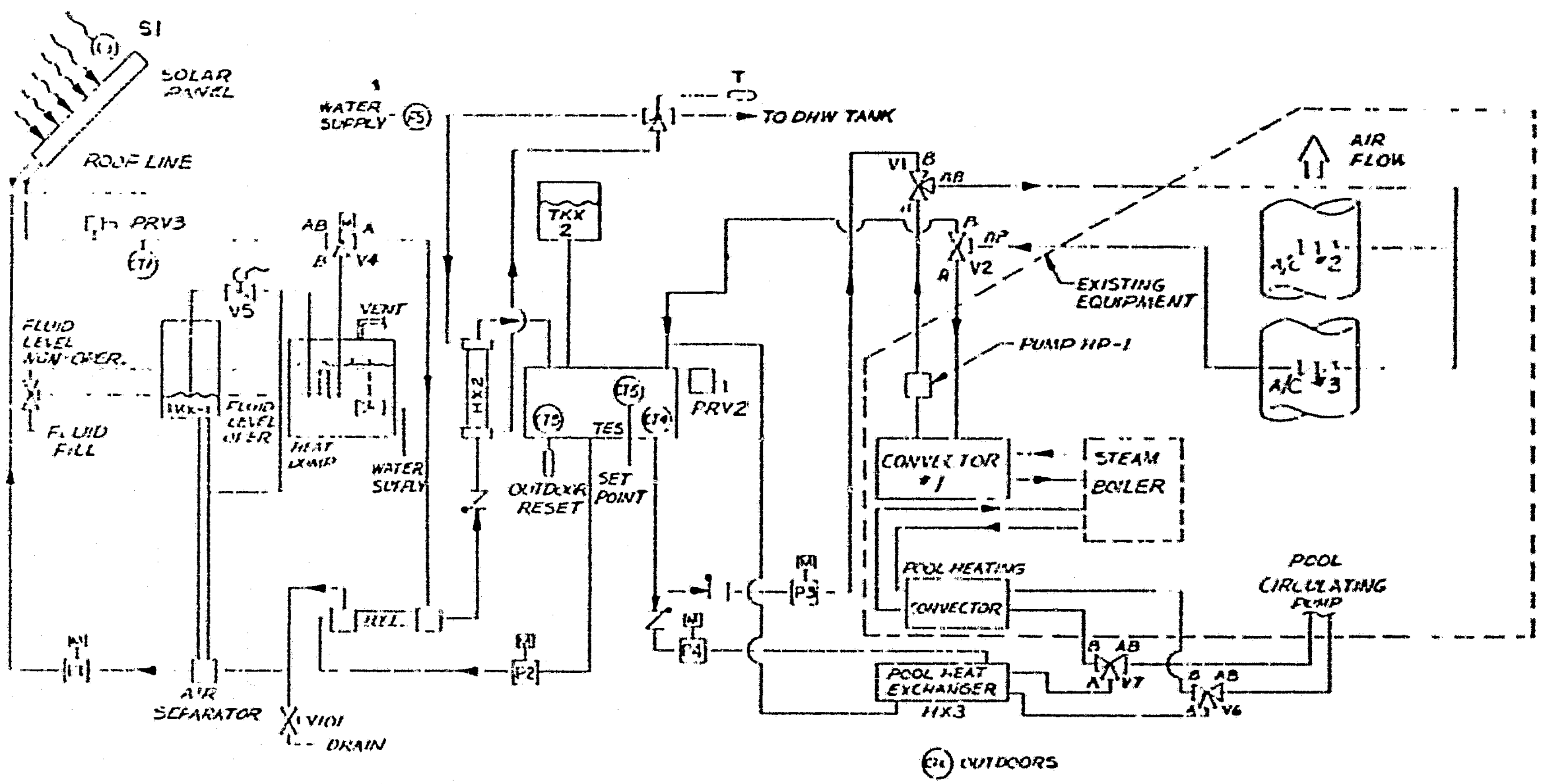


Figure 3-4. Schematic Diagram for Spokane, Washington YWCA System

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overtemperature conditions exist. The system will respond control wise as the loop II system.

3.2.5 Dallas, TX Residence

The Dallas site has the first solar cooling unit developed under this contract. The house is small and does not have a useable basement. For this reason the thermal storage tank and other system equipment are located in the garage. The existing gas furnace has been abandoned and the heat pump air handler was located in the attic. An all new duct system was designed and installed to accommodate the unit. This is not a typical retrofit for a residence. The cooling unit is normally 3 tons.

The system schematic is shown in Figure 3-5 and the size summary is contained in Table 3-3. The control system is as described in Section 2.3.6.

3.2.6 Murphy, NC and Muscle Shoals, AL TVA sites

The Murphy and Muscle Shoals sites will each receive 10 ton solar heating and cooling systems. At the close of this contract, the equipment has been delivered but the site installations have not been designed. The system schematic is shown in Figure 3-6 and the system size summary is contained in Table 3-3. Because of the low use of domestic hot water, these sites have been designed to exclude solar pre-heat of hot water. This was an application decision versus a general system recommendation. The controls for those systems are as described in Section 2.3.6.

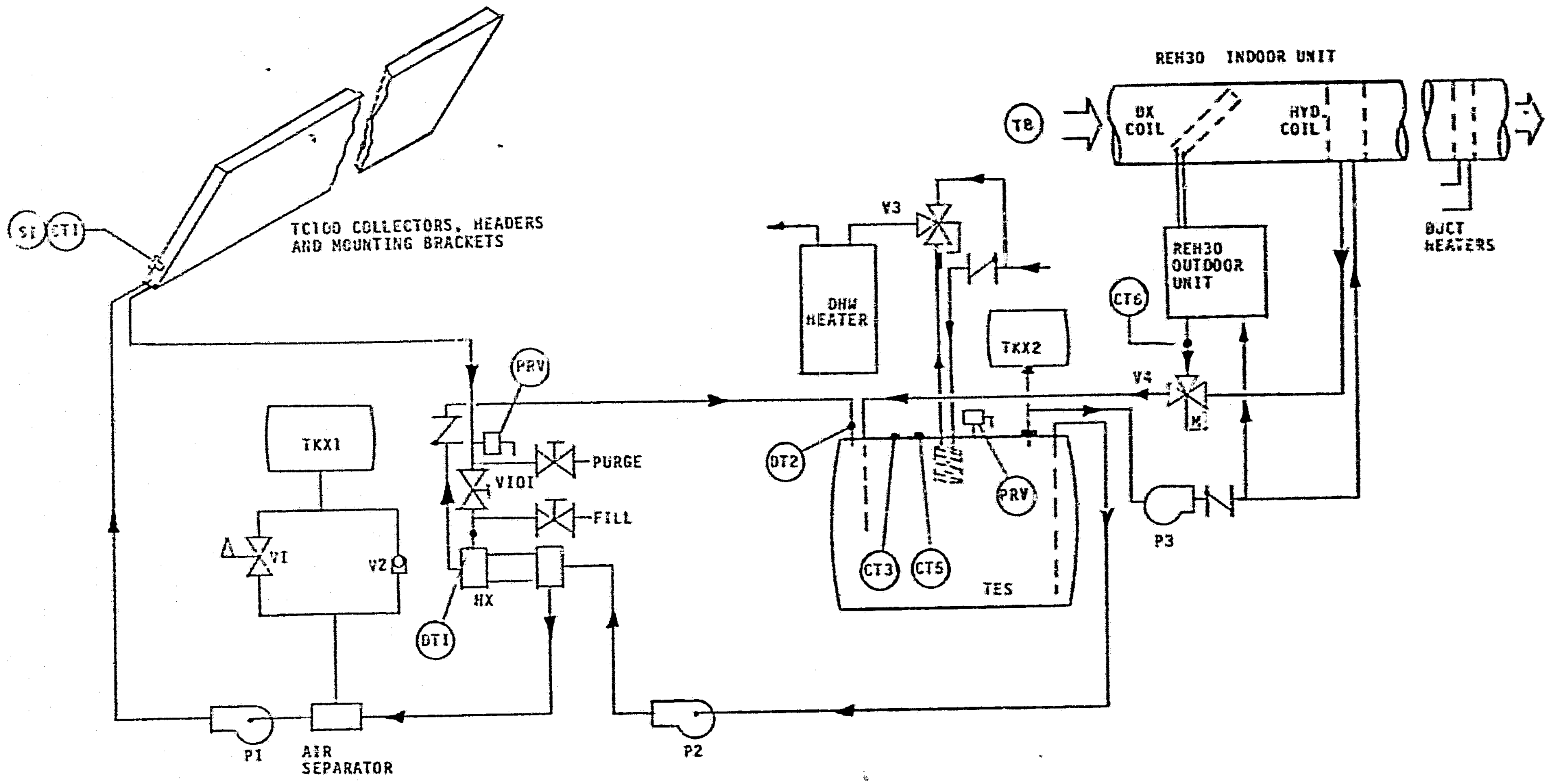


Figure 3-5. Schematic Diagram of the Solar Heating and Cooling Systems at the Dallas, Texas O.T.S.

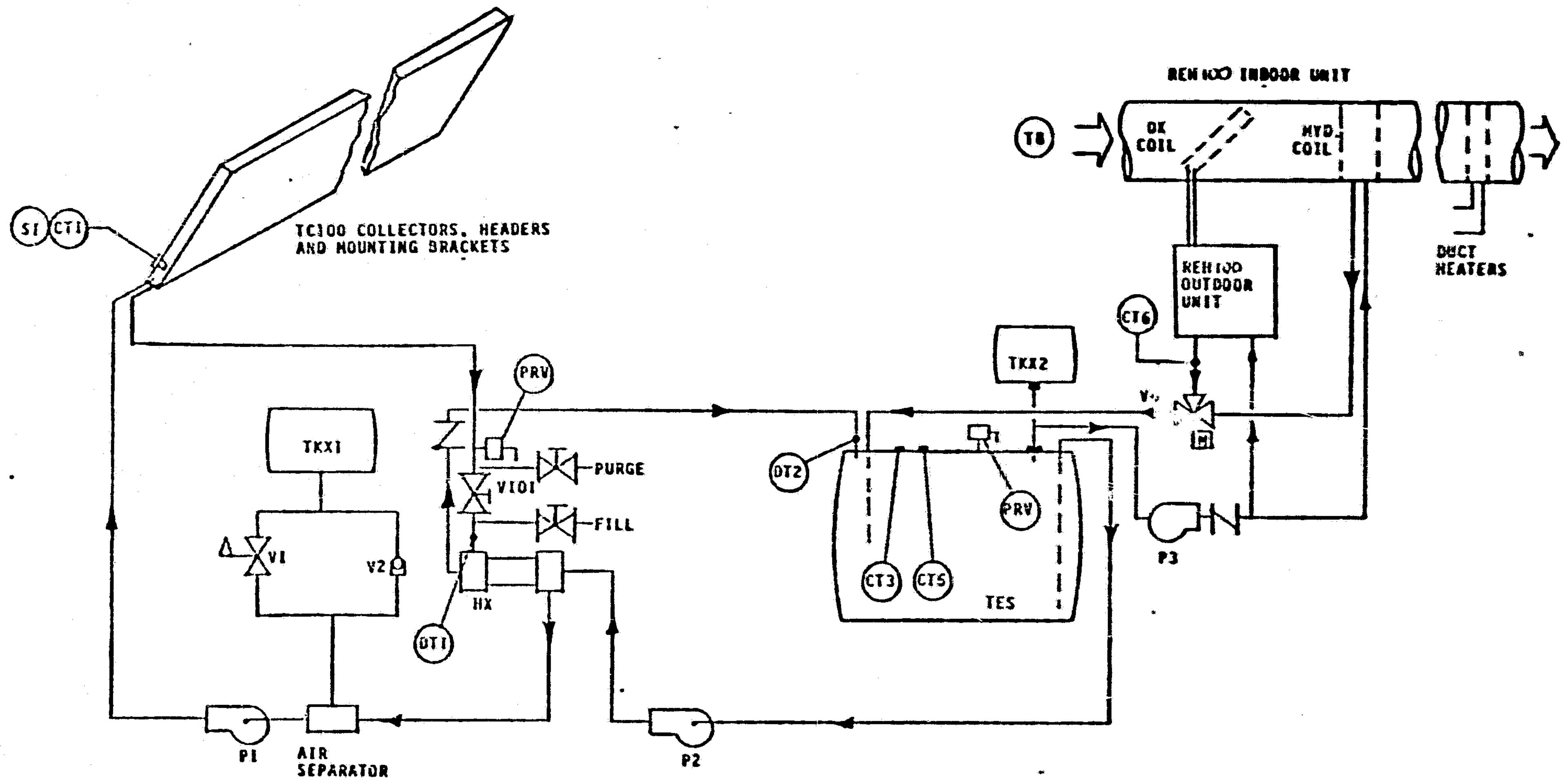


Figure 3-6. Diagram for Solar Heating and Cooling Systems, TVA Sites

4.0 Performance

4.1 Heating Systems Performance

Selected months from the monthly "Automated Data Listings" were reviewed to evaluate the performance of the heating systems. Only Normal and Spokane had gone through a full season. Ft. Meade and Milwaukee came on line at the end of the heating season and as a result do not offer full potential for a performance review. The months selected for analysis were:

Ft. Meade - May 1979 (earliest available)
Normal - March 1979
Spokane - April 1979
Milwaukee - August 1979 (latest available)

March was selected for Normal based on having a full months' worth of good data at the same time that a significant load existed. During December, January and February, the collectors became snowbound and the data received was not appropriate. Also, March represented a period after a year of operation and therefore reflects the performance beyond the "new installation" time period.

The Spokane site was also effected by heavy snowfalls and therefore like Normal, December, January and February did not produce appropriate data. Some work by a site contractor to remove a leaking heat exchanger in the swimming pool loop caused the system to remain shut down for the first nine days in March. April was used since the system was operating all month and a significant heat load was present.

4.1.1 Collector Loop Efficiencies

Table 4.1-1 contains data and predictions of the collector area of 17.8 ft^2 per panel. The measured performance (subject to normal accuracy of site data systems) is shown both as the maximum value seen for any one day and the average for the month. The design value is a monthly average as obtained from a computer simulation of the site for the same month. The design was completed prior to having collector data taken by DSET, Inc. The expected average is based on the collector performance using data obtained at DSET, Inc.

The daily average expected efficiency was determined by computing the all day daily efficiency using the noon time efficiency and the incident angle modifier along with the clear day insolation for a 40° tilt angle collector located at 40° latitude. The noon time efficiency based on a 14.8 ft^2 net area fit the following equation:

$$\eta = 0.613 - .159 \frac{T_f - T_a}{I} - .0868 \frac{\sigma (T_f^4 - T_a^4)}{I}$$

and the curve of $K_{\alpha\tau}$ is shown in Figure 4-1. The result of all day efficiency calculations was a fit to the following equation:

$$\eta_{\substack{\text{all}\\\text{day}}} = 0.496 - 2.70 \frac{\Delta T}{I_{\substack{\text{avg.}\\\text{daily}}}}$$

Since the data measures collector performance and pipe losses, an additional 5% loss factor is expected. With the loss factor and converting the equation to reflect the gross area of 17.8 ft^2 used in NASA ADL, the all day efficiency equation becomes:

$$\eta_{\substack{\text{all}\\\text{day}}} = 0.412 - 2.362 \frac{\Delta T}{I_{\substack{\text{avg}\\\text{daily}}}}$$

Table 4.1-1

Collector Array Efficiencies

SITE	MONTH	DAILY EFFICIENCIES ACTUAL MAX	ACTUAL AVG	DESIGN AVG	EXPECTED AVG
Normal	March '79	.318	.216	.341	.251
Spokane	April '79	.351	.327	.341	.308
Ft. Meade	May '79	.363	.176	.11	.192
Milwaukee	August '79	.230	.134	.05	.197

Incidence Angle Modifier
for the TC100 Collector
determined by DSET, Inc.

Report No. 7750317
DSET No. 18727S
4/6/78

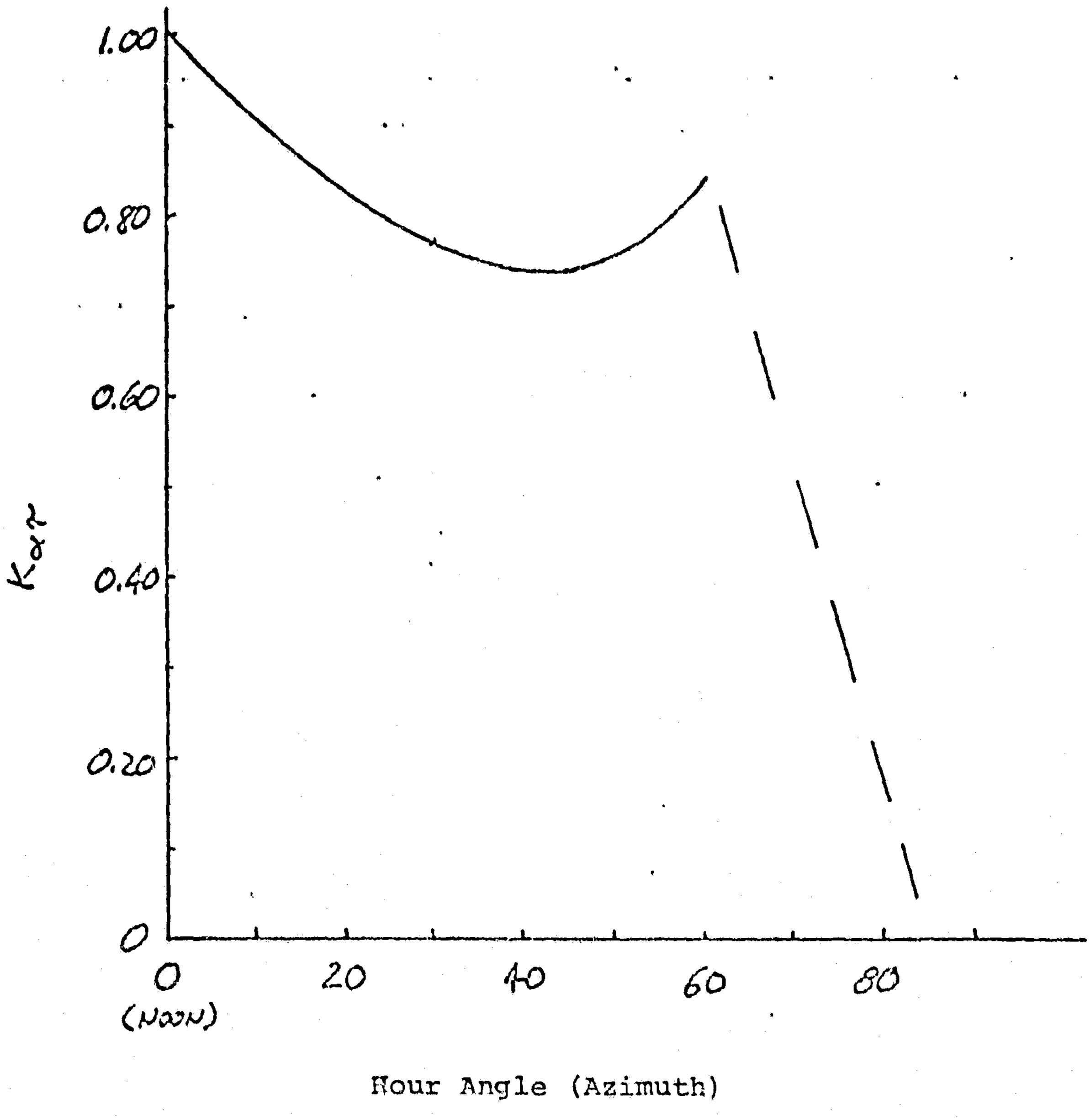


Figure 4-1

The Milwaukee site has Lexan covers and therefore will reduce the insolation by 15% reducing the equation to:

$$\eta_{\substack{\text{all}\\\text{day}}\ \text{Milwaukee}} = 0.3502 - 2.362\frac{\Delta T}{I_{\substack{\text{avg}\\\text{daily}}}}$$

A comparison between the Actual Average vs the Expected Average collector array efficiencies shows that the collector array is performing reasonably well. The Spokane data shows higher efficiencies than expected whereas the other sites show lower efficiencies. Overall the system when operating under a load such as the Spokane and Normal sites are operating near expectations.

Data accuracy is always questionable, however. As an example, at the Normal, Il. site the energy transferred to storage is higher than that collected which is clearly a problem with the data system. If the energy to storage were to be used as representative of the collector array energy the following values would be obtained:

Maximum collector array efficiency	0.336
Actual Average	0.238
Expected Average	0.251 (From Table 4-1)

As can be seen the correlation has improved considerably. The comparison between the actual and expected values changed from 86% to 95%.

4.1.2 Analysis of Normal Site

The site performance can be characterized by the energy collected and the energy delivered to the load. Table 4.1-2 contains the overall energy balances for the site compared with the design predictions and the expected results. The space heating load will not be found in the ADL for a number of reasons including:

1) The load is measured using the temperature difference across the furnace on the air side. This does not take into account the energy lost from the tank that gets to the living space by virtue of the tank and piping being indoors.

2) Residential heating loads tend to follow degree days trends closely. Historic data was available during the system design period so as to obtain the load vs degree days for that house. The load obtained in this fashion was considerably higher than from that reported in the ADL. This latter load was used in Table 4.1-2.

The biggest difference between the as designed case and as measured is in the insolation on the collectors.

Also the hot water contribution is lower than expected. The approach used is that found in hydronic heating systems. This approach will cause the heater to activate if the pipe length between the coil in the TES and the DHW tank is long (and in Normal there is 15' of pipe) since cool water in the pipe will enter the tank. The true energy saving potential of this approach requires large draws of water such as clothes washing and showers.

Table 4.1-2

System Performance, Normal, Il.

	ACTUAL	DESIGN	EXPECTED
Insolation	9.529	16.615	9.529
Load H	12.80	12.90	12.80
DHW	1.76	2.212	1.76
Energy Collected	2.265	4.697	2.392
% Collected	23.8	28.3	25.1
To DHW	0.422	1.88	0.422
%	24	85	24
To Heat	1.843	2.817	1.970
% H	14.4	21.8	15.4

NOTE: All values in Millions of BTUs.

4.1.3 Analysis of Spokane Site

Table 4.1-3 contains the overall energy balances from the site compared with the design predictions. The expected results are the same as the actuals since the expected collector loop performance is very close to the actual performance.

The values taken from the ADL in Table 4.1-3 are the insolation, the heating load, the domestic hot water (DHW) load, the energy collected, and the energy transferred to the DHW load. There is a significant difference between energy collected and energy stored, and also between energy stored and energy used. The system has high flow rates in the storage and distribution loops and therefore, the temperature differences vary between 0 and 5°F.

The losses from the system were computed using actual pipe lengths and 1/2 the insulation thickness of that installed on the system assuming that the system was at the mean storage tank temperature for 24 hours/day for the entire month. The losses computed were:

Storage loop (1500' piping)	=	8,471,520 BTU/Month
TES tank (685 ft^2)	=	2,564,640 BTU/Month
Distribution loop (800' piping)	=	4,528,800 BTU/Month

Using these conservative estimates for the energy losses the energy transferred to the storage tank would be

$$\text{Energy to Storage} = \text{energy collected} - \text{DHW load} - \text{storage loop losses}$$

or

$$\text{energy to storage} = 60.2 - 6.8 - 8.7$$
$$= 44.7 \text{ million BTUs}$$

The energy transmitted to the heating load would then be

$$\text{energy to Heating System} = \text{Energy in Storage} - \text{TES tank losses} - \text{Dist. loop losses}$$

or

$$\text{energy for Heating System} = 44.7 - 2.6 - 4.5$$
$$= 37.6 \text{ million BTUs}$$

The total losses in the storage and distribution systems of 15.8 million BTUs represent 26% of the collected energy, a rather high value. This high value is a function of the very large amount of large piping that exists in this installation, a consequence of a retrofit installation.

The design values show considerably higher heating and DHW loads and energy collection. Once again, this is a significant difference in the actual insolation vs the predicted insolation. Since the collection efficiency is as expected however, the major difference between design and actual performance lies in the magnitude of the values. The design heat loss was 9.9 million BTUs and this is lower than the conservative estimate used in analyzing the data.

Overall, this site is operating as well as it can and the performance is limited by the insolation on the collectors.

Table 4.1-3

System Performance, Spokane, Wa.

		ACTUAL	DESIGN
Insolation		184.2	296.5
Load	Heating	38.1	137.5
	DHW	20.0	54.2
Energy Collected		60.2	93.7
% Collected		32.7	31.6
To DHW		6.8	41.7
& DHW		34.0	77.0
To Heating System		37.6	42.1
% Heating		98.0	30.6

NOTE: All values are in Million BTUs.

4.2 Rankine Electric Heat Pump Performance

4.2.1 REH-30 Performance

Tables 4.2-I and 4.2-II present design point test results for the REH-30.* Table 4.2-I shows predicted results versus measured results in the Rankine engine driven mode. The primary deficiency of the REH-30 compared to predictions is that of the Rankine engine expander. Expander efficiency and shaft horsepower are considerably below predicted values. Significant improvement in expander performance was achieved for the REH-100 and REH-101 expanders as will be shown in paragraph 4.2.2.

Figures 4.2-1 through 4.2-4 present design and off-design performance for the REH-30. As shown, overall performance is quite good for hot water temperatures above 275°F and for outdoor ambient temperatures below 95°F. Due to compressor bearing speed limitations and due to compressor high oscillating torque excursions, the minimum speed at which the REH-30 is allowed to run is 1200-1300 RPM. This imposed a severe penalty on off-design performance and will restrict seasonal solar cooling operating time. Again this deficiency has been corrected for the REH-100 and REH-101.

*REH-30 is a 3-ton unit.

4.2.2 REH-100/101 Performance

The REH-100 and REH-101 provide 10 tons net cooling capacity at design conditions.

Design Conditions

Hot Water Inlet Temp	-	$300^{\circ}F$
Outdoor Ambient Temp	-	$95^{\circ}F$ d.b.
Indoor Ambient Temp	-	$80^{\circ}F$ d.b.
	-	$67^{\circ}F$ w.b.

As will be shown in the following tables and Figures, the REH-101 has higher performance characteristics since it contains an updated expander with higher efficiency and other modifications to the Rankine engine and heat pump to reduce internal losses.

Tables 4.2-III and 4.2-IV present the design point performance of the REH-100 in the Rankine driven and electric driven mode respectively. Tables 4.2-V and 4.2 VI present the design point performance of the REH-101 in the Rankine driven and electric driven mode.

Figures 4.2-5 through 4.2-8 present the off-design performance of the REH-100 and REH-101. As shown the REH-101 is a better performer than REH-100 for a $95^{\circ}F$ outdoor ambient temperature, but the reverse is true for an $85^{\circ}F$ outdoor ambient. This should not be the case. The poor performance of REH-101 for $85^{\circ}F$ ambient is believed to be associated with the Rankine engine controls, but time was not available to investigate this problem while the unit was in the test cell.

As shown, the REH-100/101 have good performance for hot water inlet temperature above 255°F and for ambient temperatures below 95°F. The low speed cut-off for these units is 900 RPM as compared to 1200 RPM for the REH-30. It should be noted that peak $OCOP_{th}$, Figure 4.2-6, occurs at a water inlet temperature below the design point temperature of 300°F. This is due primarily to three things: (1) expander speed decreases, and as speed decreases expander efficiency increases, (2) evaporator temperature increases, (3) condenser temperature decreases. All three tend to increase OCOP for a time but eventually Rankine cycle efficiency and heat pump capacity decreases at a higher rate and an overall decay in performance results.

For a solar heated air conditioning system where design point hot water temperatures may not be achieved except for short intervals, it is necessary to direct more attention to improving off-design performance of the REH. On the other hand, it is important to select an appropriate REH design point which is consistent with the solar collector/TES thermal performance so as not to cause full time operations at an unfavorable off-design condition.

Laboratory tests have consistently shown that the Rankine Engine/Compressor are at or above design speed in less than 30 seconds at design point ambient and hot water conditions. The REH is essentially stabilized at peak cooling in approximately two minutes. Rankine engine heat losses have been measured to be less than 1 1/2% of heat input.

TABLE 4.2-I

REH-30 PERFORMANCE AT DESIGN POINT (RANKINE DRIVEN)

	INITIAL PREDICTION	UPDATED PREDICTIONS 1/1/79	TEST RESULTS* 4/3/79
o NET COOLING CAPACITY	36,000 BTUH	36,750 BTUH	(31,500 BTUH)
o COMP. EFF.	76%	(83%)	(80.6%)
o HP COP (MECH.)	5.0	(5.54)	(5.38)
o EXPANDER EFF.	82.6%	82.6%	(61%)
o MAG. DRIVE EFF.	96%	(99%)	(99%)
o OCOP	.70	.75	(.59)
o SOLAR EER	20	22.5	(20.5)

( ) MEASURED VALUES
* 95^{o}F O.D. AMB.
80^{o}F DRY BULB
68^{o}F WET BULB
1350 SCFM

TABLE 4.2-II

REH-30 PERFORMANCE (ELECTRIC MOTOR DRIVEN) *

	95°F O.D. AMB.		90°F O.D. AMB	82°F O.D. AMB
	68°F W.B.	67°F W.B.	67°F W.B.	67°F W.B.
o NET COOLING CAPACITY	36,704 BTUH	36,883	37,100	37,882
o EER	11.25	11.48	11.83	12.69
o SENSIBLE HEAT FACTOR	0.70	0.73	0.73	0.72
o TOTAL WATTS	3260	3212	3136	2984
o EVAPORATOR TEMP, °F	47.8	47.1	46	45

* 80°F DRY BULB
1750 RPM

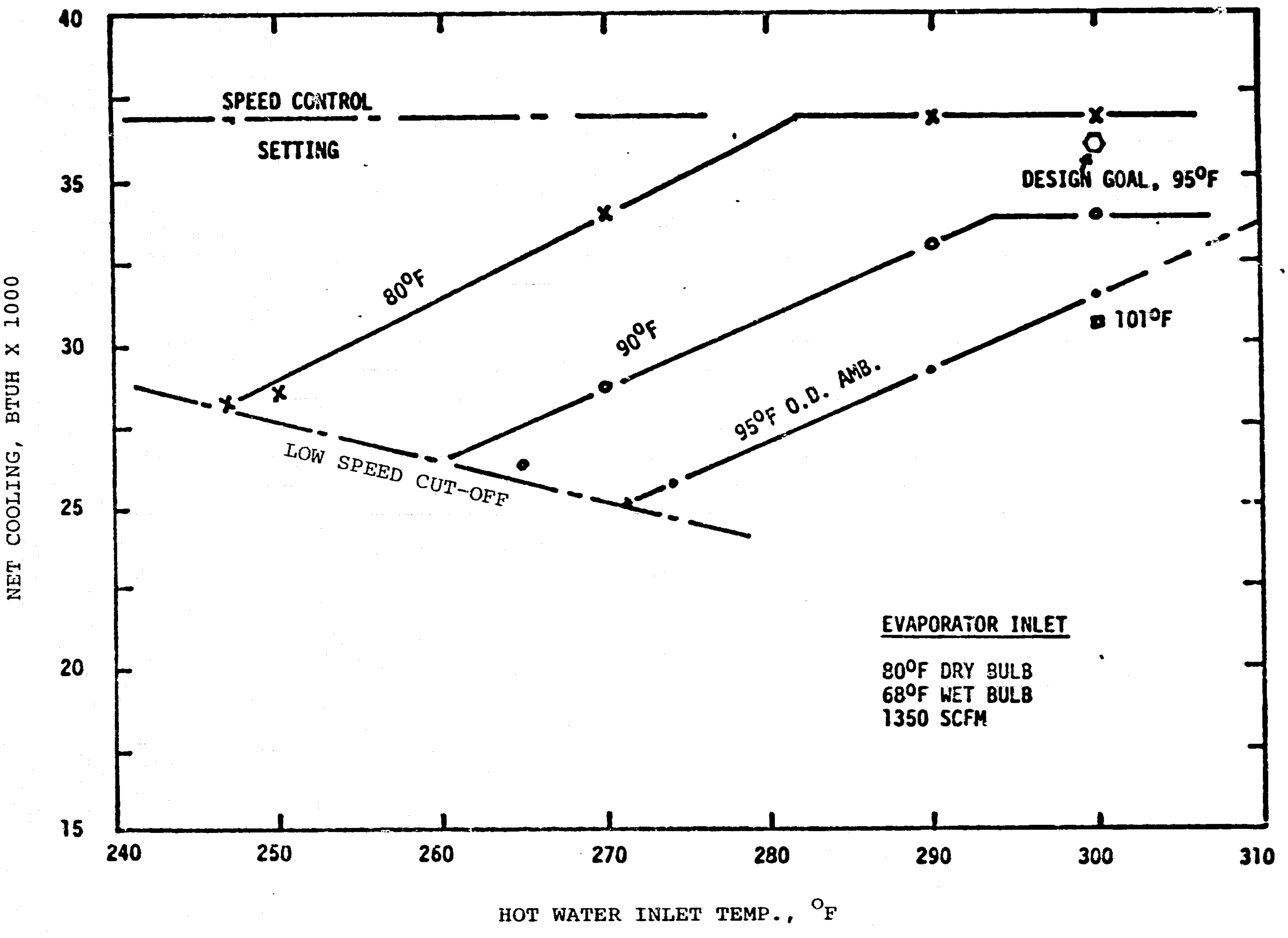


FIGURE 4.2-1 REH-30 Rankine Driven Net Cooling Effect

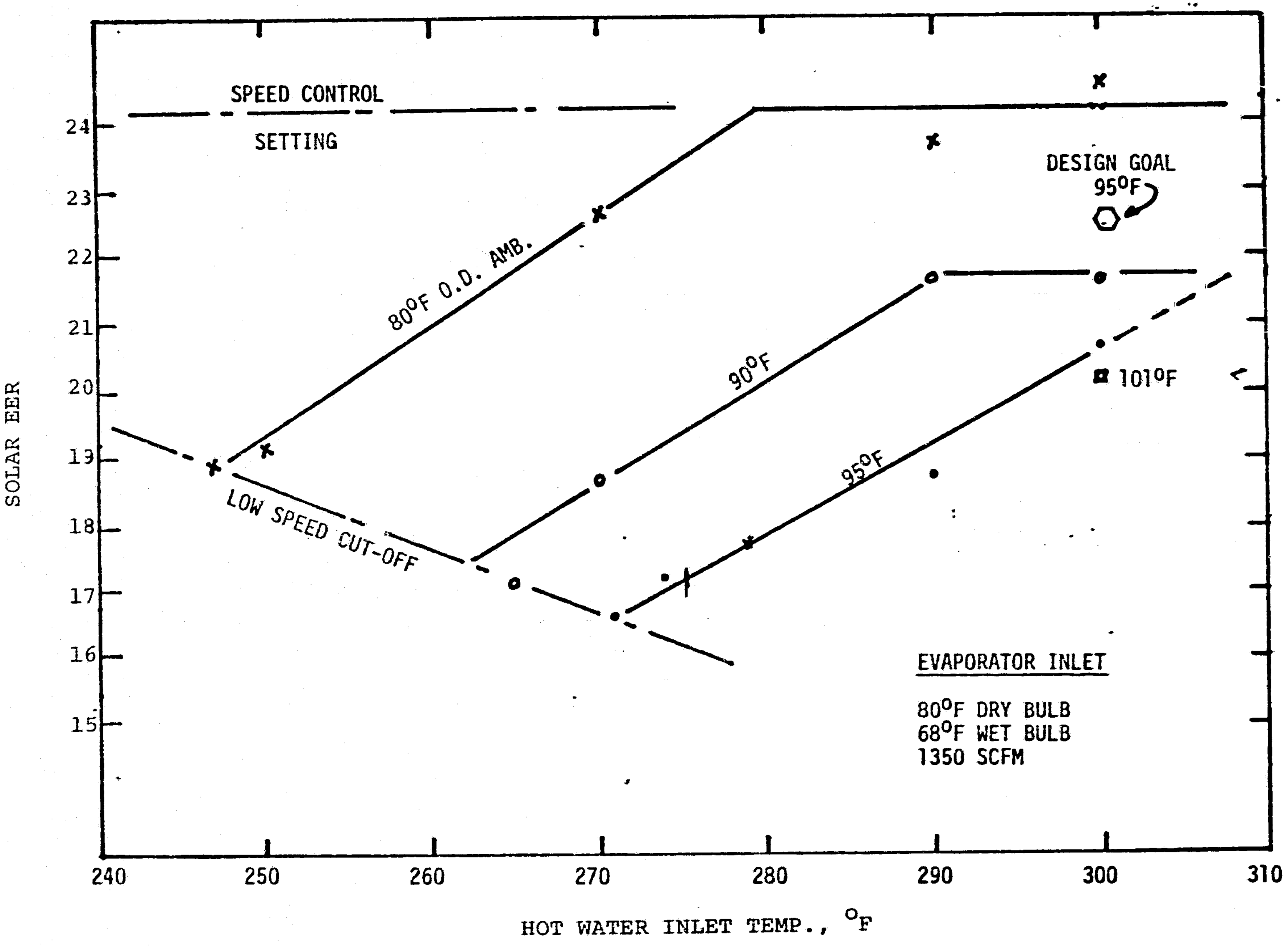


FIGURE 4.2-2 REH-30 Rankine Driven EER

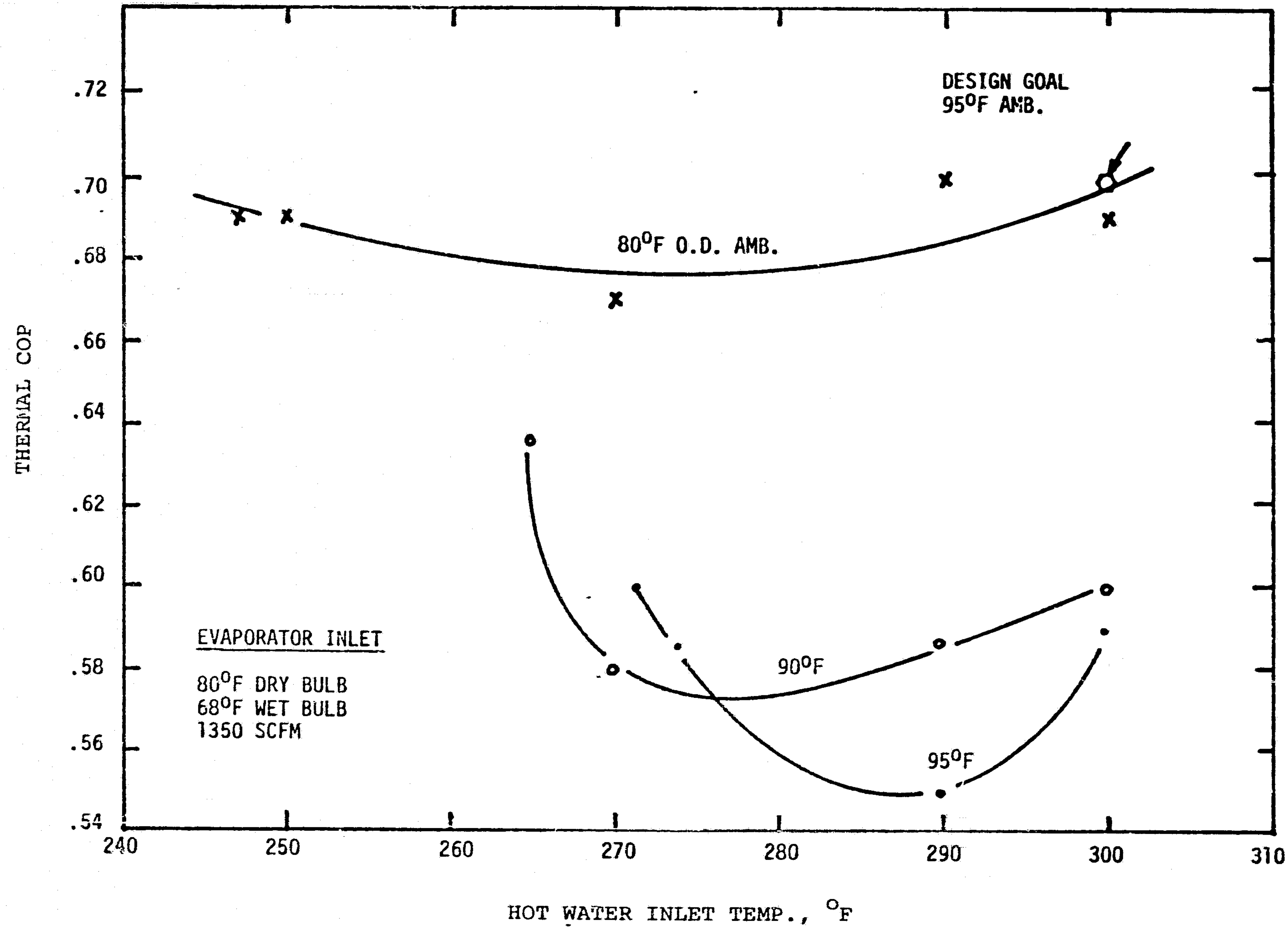


FIGURE 4.2-3 REH-30 Overall Thermal COP

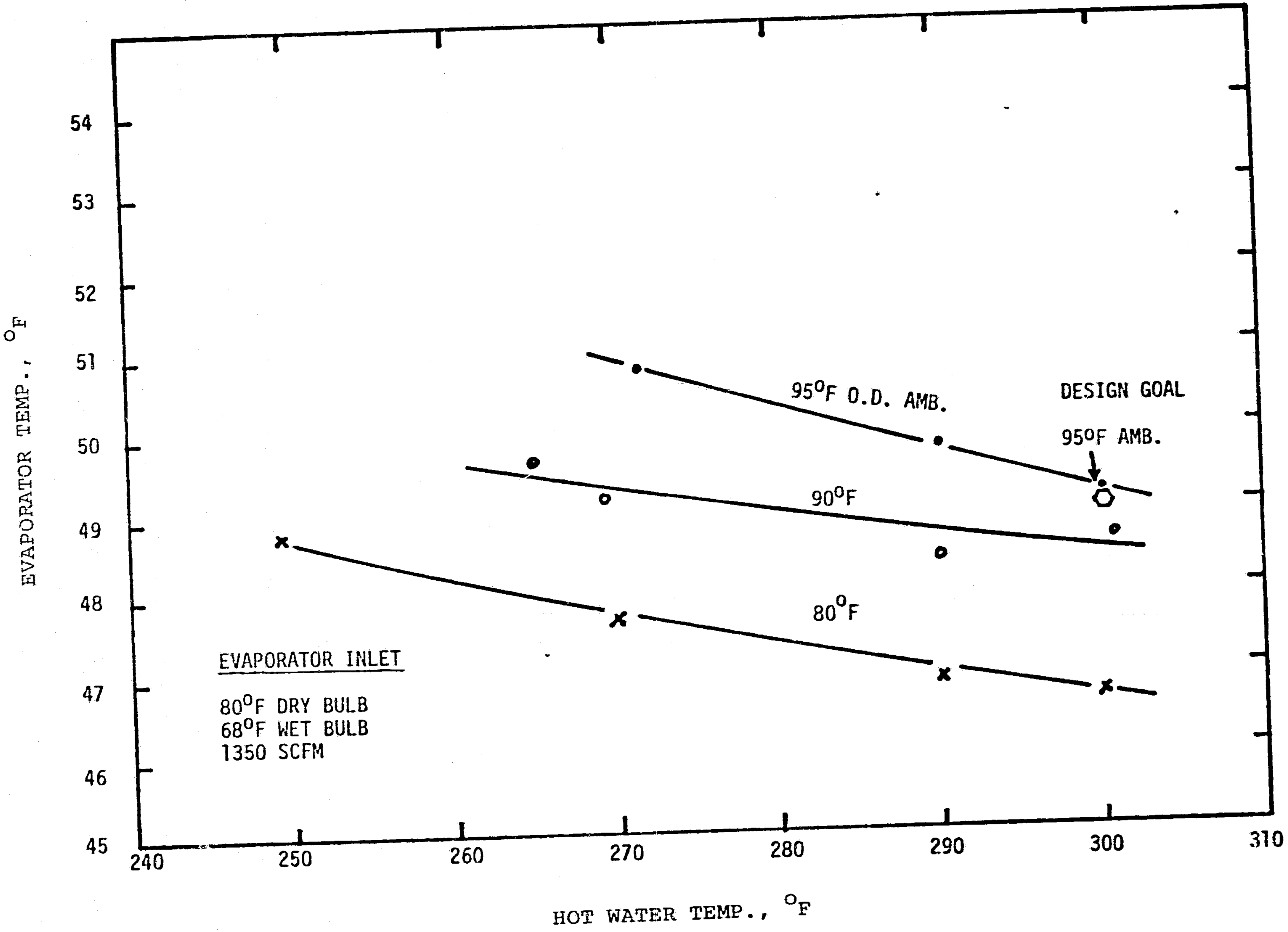


FIGURE 4.2-4 REH-30 Heat Pump Evaporator Temp. (Rankine Driven)

TABLE 4.2-III

REH-100 DESIGN POINT PERFORMANCE (RANKINE DRIVEN)

	TEST RESULTS	DESIGN GOAL
o NET COOLING CAP., BTUH	116,573	120,000
o EER	19.5	20
o $OCOP_{th}$	0.60	.74
o SENSIBLE HEAT FACTOR	.74	--
o EVAP. TEMP., ^{o}F	49	49
o CONDENSING TEMP., ^{o}F	112	110
o TOTAL WATTS	5974	6000

TABLE 4.2-IV

REH-100 DESIGN POINT PERFORMANCE (ELECTRIC DRIVEN)

o NET COOLING CAP.	119,212 BTUH
o EER	12.2
o SENSIBLE HEAT FACTOR	.74
o EVAPORATOR TEMP., $^{\circ}F$	48.5
o CONDENSING TEMP., $^{\circ}F$	107
o TOTAL WATTS	9771

TABLE 4.2-V

REH-101 RANKINE PERFORMANCE

	TEST RESULTS	DESIGN GOAL
o NET COOLING CAP., BTUH	126,456	120,000
o EER	20.65	20
o $OCOP_{th}$	.68*	.74
o SENSIBLE HEAT FACTOR	.74	---
o EVAP. TEMP., ^{o}F	49.1	49
o CONDENSING TEMP., ^{o}F		
- Rankine	109	110
- Heat Pump	111.5	110
o TOTAL WATTS	6123	6000

* $OCOP_{th}$ = .73 for 275^{o}F Water

TABLE 4.2-VI

REH-101 ELECTRIC PERFORMANCE

o NET COOLING CAPACITY	125,224 BTUH (10.43 TONS)
o EER	12.88
o SENSIBLE HEAT FACTOR	.77
o EVAPORATOR TEMP., ^{o}F	48
o CONDENSER TEMP., ^{o}F	108.5
o TOTAL WATTS	9722

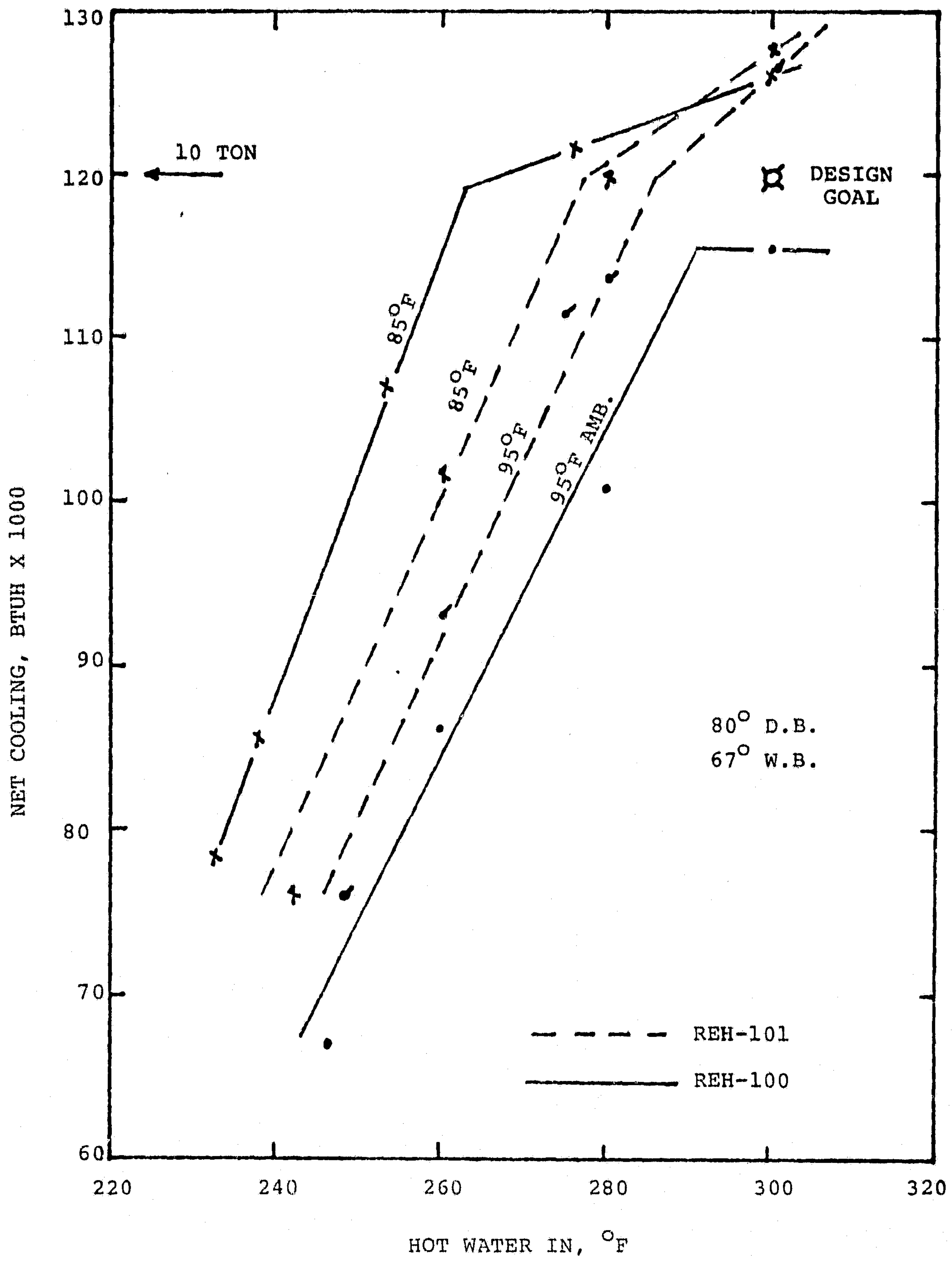


FIGURE 4.2-5 REH-100/101 Rankine Driven Net Cooling Effect

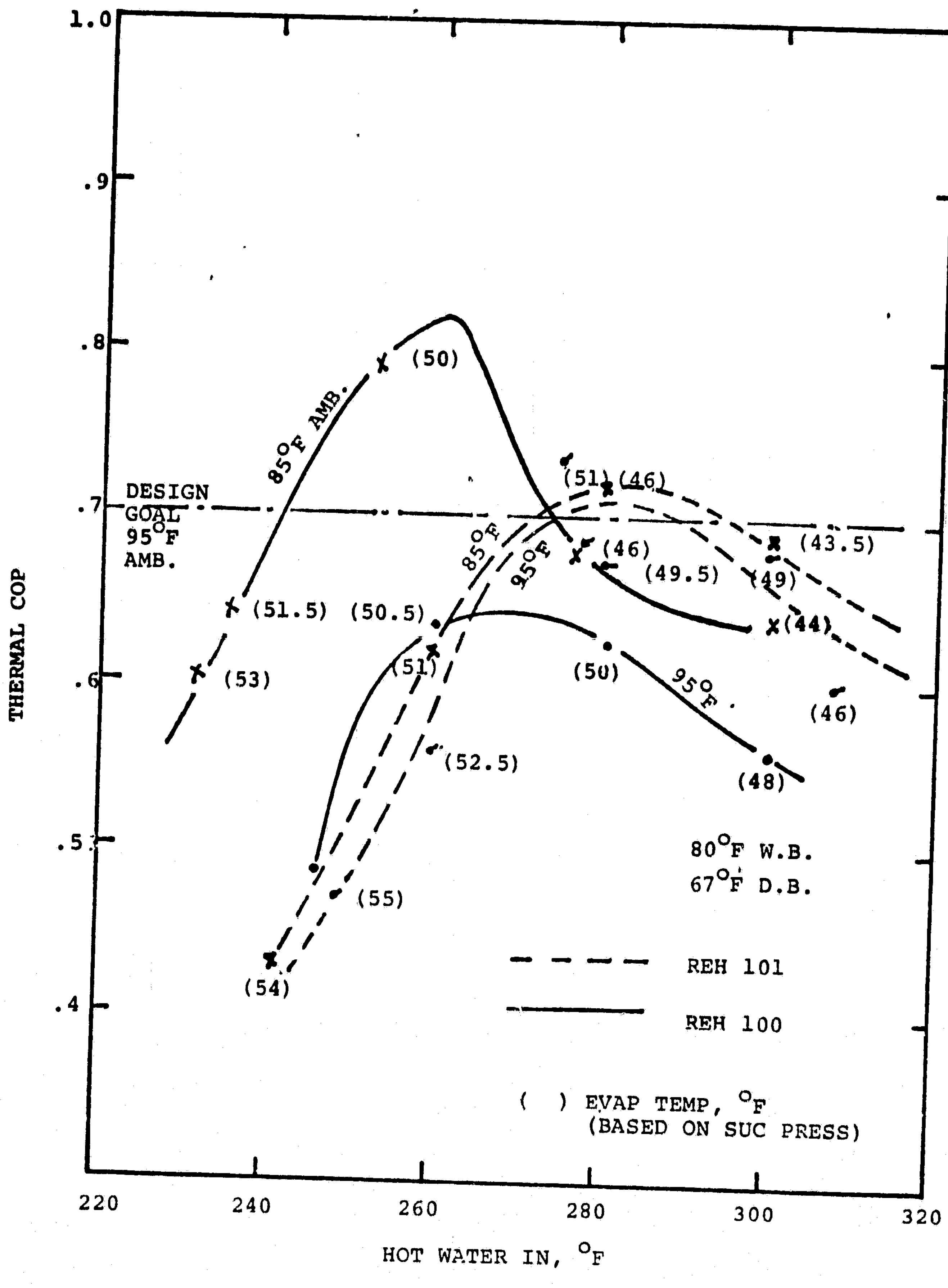
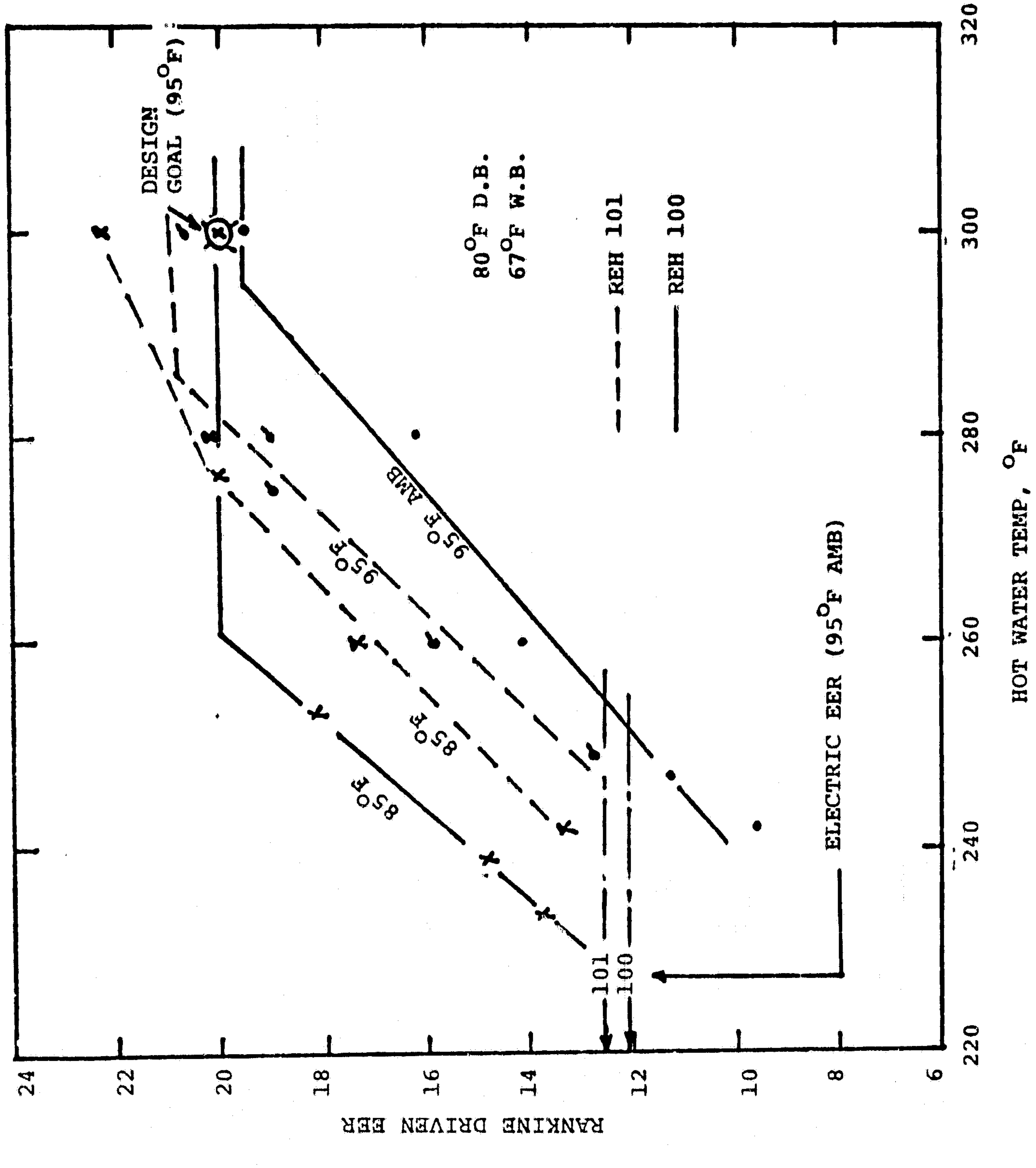


FIGURE 4.2-6 REH-100/101 Rankine Driven $OCOP_{th}$



DESIGN GOAL (95^{O}F)
80^{O}F D.B.
67^{O}F W.B.
REH 101
REH 100
95^{O}F AMB
95^{O}F
85^{O}F
85^{O}F
ELECTRIC EER (95^{O}F AMB)
101
100
RANKINE DRIVEN EER
HOT WATER TEMP, ^{O}F
24
22
20
18
16
14
12
10
8
6
220
240
260
280
300
320

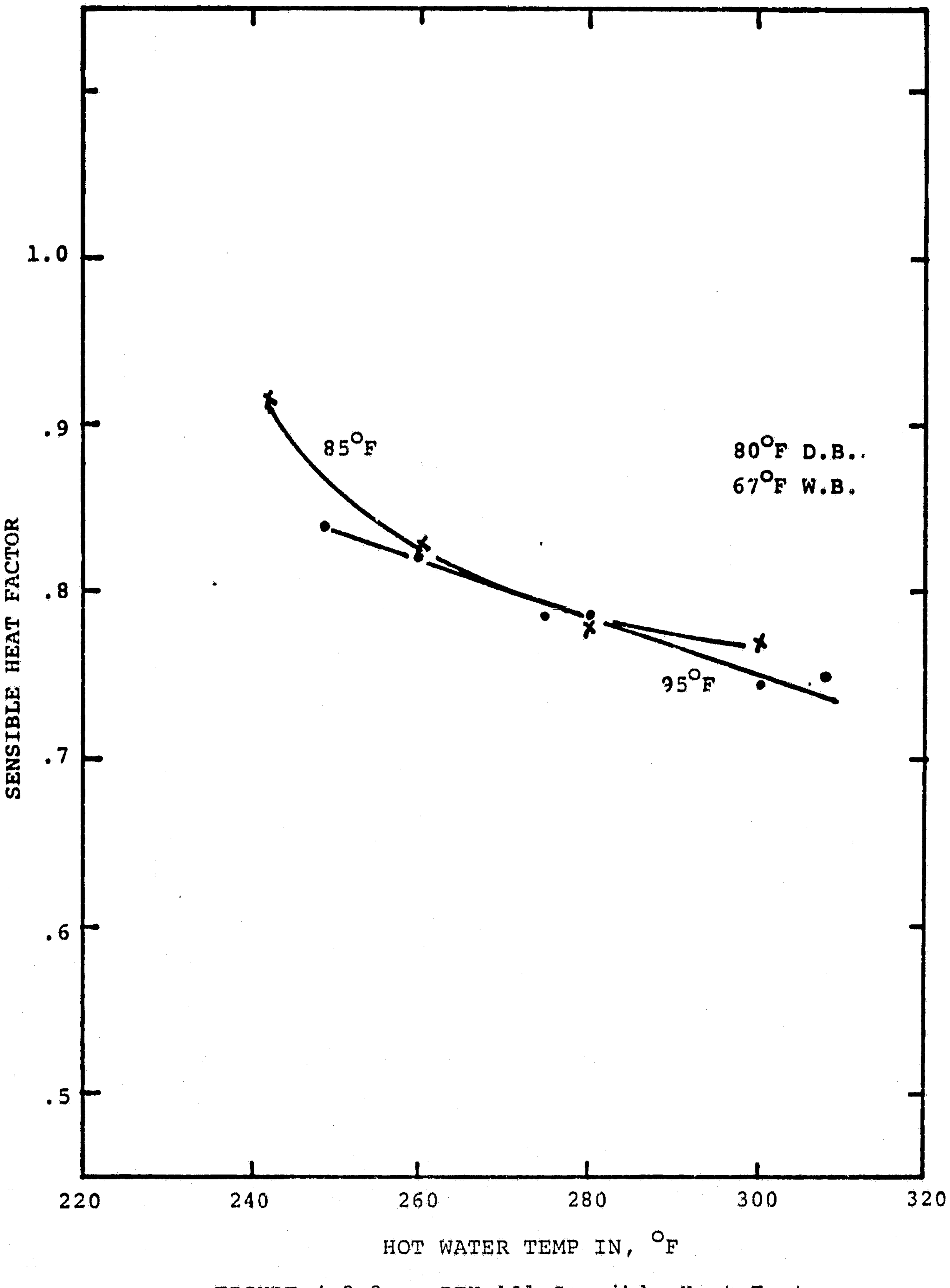


FIGURE 4.2-8 REH-101 Sensible Heat Factor

4.3 ECONOMICS

Due to a number of site problems caused by component malfunctions, contractor and owner caused problems, there was insufficient data from which to establish the total energy savings potential for the solar heating systems. However, since the collectors are operating close to expectations it is reasonable to assume that energy saving studies will prove to be good approximations of the future potential for solar heating systems after

a) hardware bugs are removed

b) Installer and servicemen are more familiar with the equipment

c) Building owners understand the system controls interactions

To this end, a look at recent economic studies can give insight to the potential energy savings of solar heating and cooling systems.

4.3.1 APPROACH

Levelized Annual Cost - True life cycle cost analysis must necessarily consider the timing of costs and benefits as well as the magnitude. A method employed in previous General Electric solar and wind energy programs is to compare Levelized Annual Benefits (LAB), representing system energy savings, with the Levelized Annual Cost (LAC), the levelized dollar amount required to own, operate, and maintain a system during each year of the life of the system. Specifically, the levelized annual cost accounts for:

1. "Paying off" system capital costs

2. Paying for operating and maintenance expenses, including auxiliary energy

3. Paying taxes

4. Paying a return to investors and interest to creditors

5. Building a capital fund for periodic component replacement, overhaul, and retirement of debt.

The levelized annual cost, denoted by LAC, is given by:

$$LAC = CRF \times PV$$

where CRF is the capital recovery factor and PV is the present value of the year by year revenue requirements throughout system life.

The present value is analogous to that amount which, if deposited in an interest bearing account at the discount rate, would permit annual withdrawals to pay all system costs and diminish to zero at the end of system life. The levelized annual cost is equivalent to annual deposits in the account. LAC can be computed in current dollars (as is the typical home mortgage) or in constant dollars for a particular base year. Figure 4.3-1 presents graphically the procedure for computation of levelized annual cost.

<u>Levelized Annual Benefits</u> - The comparison of the energy cost savings of the solar cooling system to the levelized annual cost is accomplished by computing the levelized annual benefits (LAB) for the energy savings. LAB is inherently a function of present

and projected energy prices and may be expressed by

$$\text{LAB (Constant \$)} = M_f \cdot P_o \cdot E$$

where E represents the annual energy saved by the solar system, M is an energy savings multiplier which accounts for the rate of energy price escalation over the lifetime of the system, and P_o is the energy price in year zero. In actual practice the appropriate utility rate schedule is applied with the savings determined by the difference of the electric bills computed with conventional and solar cooling systems.

For given price and energy savings in the base year levelized annual benefits obviously increase with energy price escalation rate and, for all rates greater than zero, with number of years to system startup. Figure 4.3-2 presents the multiplier, M, plotter versus start year and escalation rate for an assumed 20 year system life and 10 percent cost of capital, or discount rate.

4.3.2 Solar Heating and Domestic Hot Water Systems

The economics of a solar heating domestic hot water system is dependent not only on the climate and load match but also on the financial environment of the prospective user. A scenario has been developed for both residential users and commercial/industrial users to verify the cost effectiveness of solar heating and domesti (process) hot water applications. Table 4.3-1 shows two assumed typical financial environments, one for a homeowner and the other for the commercial/industrial building owner.

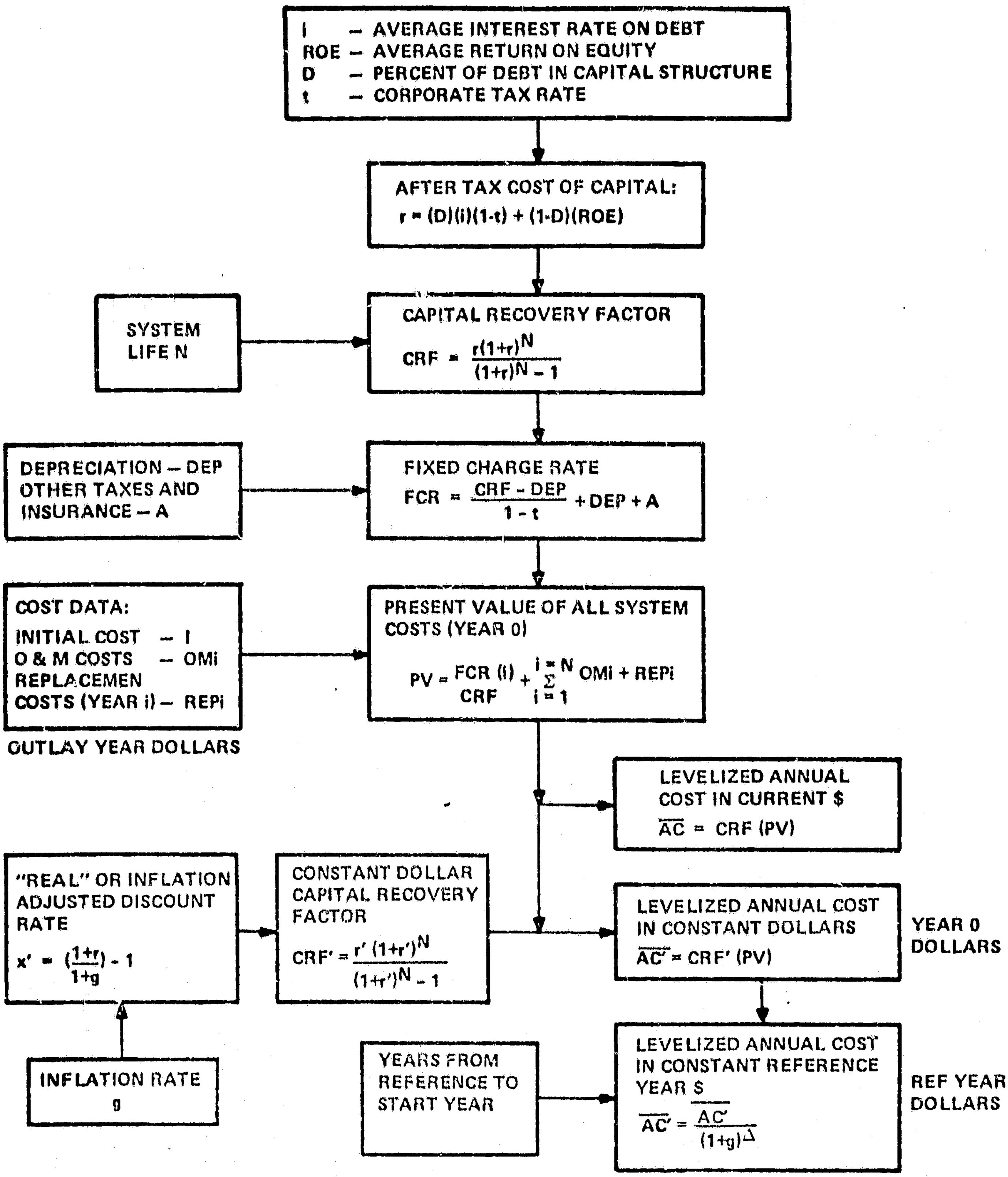


Figure 4.3-1. Levelized Annual Cost Computation

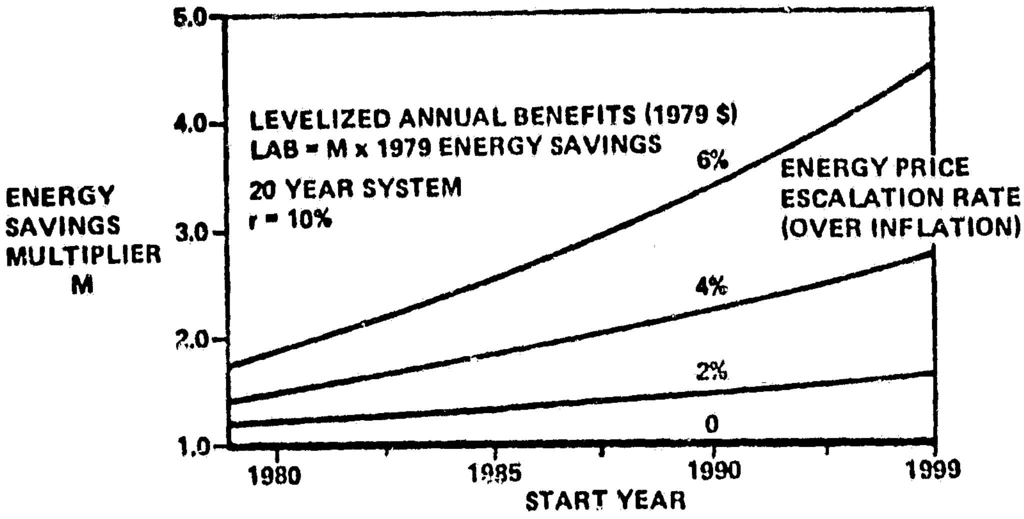


Figure 4.3-2. Energy Savings Multiplier

The economic viability of a system is measured by comparing the levelized annual cost to the levelized annual benefits. If the levelized annual benefits exceed the levelized annual cost, the system is economically viable. The break-even system cost occurs when LAC and LAB are equal. System break-even cost can be plotted versus start year and price escalation rate, yielding curves similar to those of Figure 4.3-2. System cost projections can then be cross plotted, as shown in Figure 4.3-3 to determine the conditions leading to economic viability.

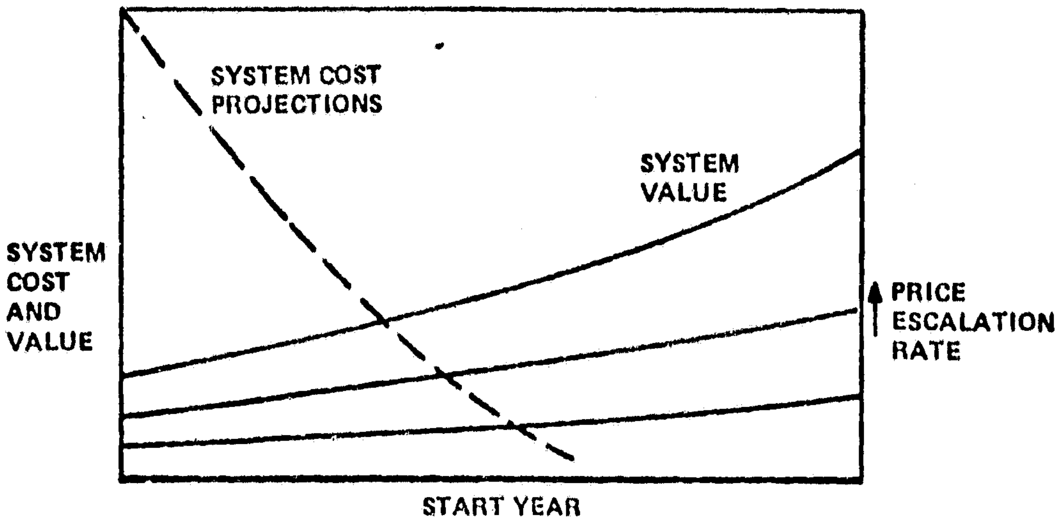


Figure 4.3-3. System Cost Versus Value

Table 4.3-1

Solar Economics Background Assumptions

HOMEOWNER	PRESENT STATUS	NEXT YEAR	ADDITIONAL CREDITS
INTEREST RATE	13%	10%	4%
INCOME TAX CREDIT	25%	25%	50%
MARGINAL INCOME TAX RATE ($25,000 INCOME)	30%	30%	30%
PROPERTY TAX (MOST AREAS)	0	0	0
INSURANCE (% OF INITIAL COST)	1/2%	1/2%	1/2%
EQUIVALENT ANNUAL PAYMENT (I.E., ANNUAL MORTGAGE RATE)	8 1/2%	7 1/2%	3 1/2%

PROCESS APPLICATION FOR INDUSTRIAL/COMMERCIAL OWNER

	PRESENT STATUS	ADDITIONAL CREDITS
INCOME TAX BRACKET	48%	48%
INVESTMENT TAX CREDIT	10%	25%
DEPRECIATION SCHEDULE	15 YRS	15 YRS
COST OF DEBT CAPITAL	10%	4%
EQUIVALENT ANNUAL PAYMENT (I.E., ANNUAL MORTGAGE RATE)	22%	15%

TABLE 4.3-2

Cost of Solar Systems

	$/Sq. Ft.
• COLLECTORS	20
• INSTALLATION	6
• STRUCTURE	6
• PIPING	6
• ENERGY STORAGE	5
• HEAT EXCHANGERS	1
• ELECTRICAL	1
	$45

The cost of solar systems have been normalized as shown in Table 4.3-2 to $45/ft^2 of collector. Here the costs are directed more to the commercial user since it includes the cost of structure. However, the costs are dependent upon a mature market and are estimates only and therefore no segregation will be made in the unit area costs between building types.

In a good solar climate, a collector module with 15 ft^2 of active area will harvest 3.5 million BTU/year. Therefore the system costs $190/million BTU/year. Because it is assumed that a loan will be used to pay the system the actual levelized cost per year is as shown in Table 4.3-3.

Table 4.3-3

Levelized Cost of Solar Energy

User	$1/Million BTU	
	Present	With Tax & Interest Incentives
Homeowner	16	7
Industrial/ Commercial	42	29

The cost of solar energy as a function of the equivalent annual mortgage rate is shown in Figure 4.3-4.

Figure 4.3-4 also contains the current cost of conventional energy. However in considering the cost of energy over the lifetime of the system, the Levelized Annual Benefits (LAB) must be considered. The LAB is defined below:

Levelized Annual Benefits = Present Value of Energy Savings x Capital Recovery Factor

$$LAB = \frac{r(1+g)}{r-g}\left[\frac{(1+r)^n-(1+g)^n}{(1+r)^n-1}\right] \times S_o$$

where r = after tax cost of capital
g = annual escalation factor for energy cost
n = system life
S_o = energy cost savings in first year

or $LAB = M \times S_o$

where M = multiplier applied to first year's energy cost savings to yield savings over system's life

Figure 4.3-5 shows the multiplier of the first year energy cost savings as a function of the energy price escalation rate. It should be noticed that with interest rates of 10% and a 10% energy escalation rate, the average energy cost is twice as high as the first year's cost. On that basis the cost comparison between solar energy and conventional energy is restructured as shown in Figure 4.3-6. Therefore the economics of solar heating and domestic hot water systems become viable based on the

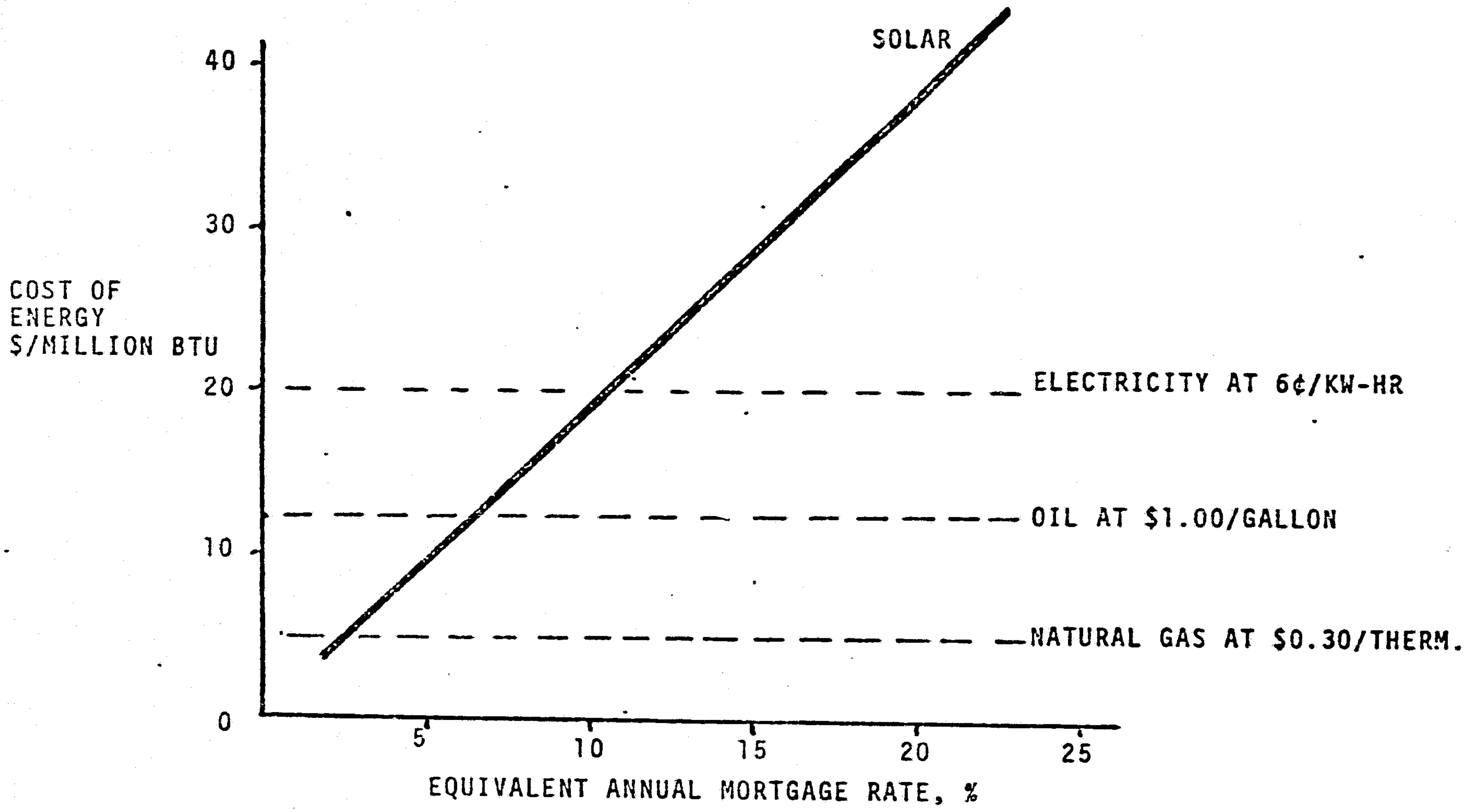


Figure 4.3-4. Comparing Solar and Non-Solar Energy Costs

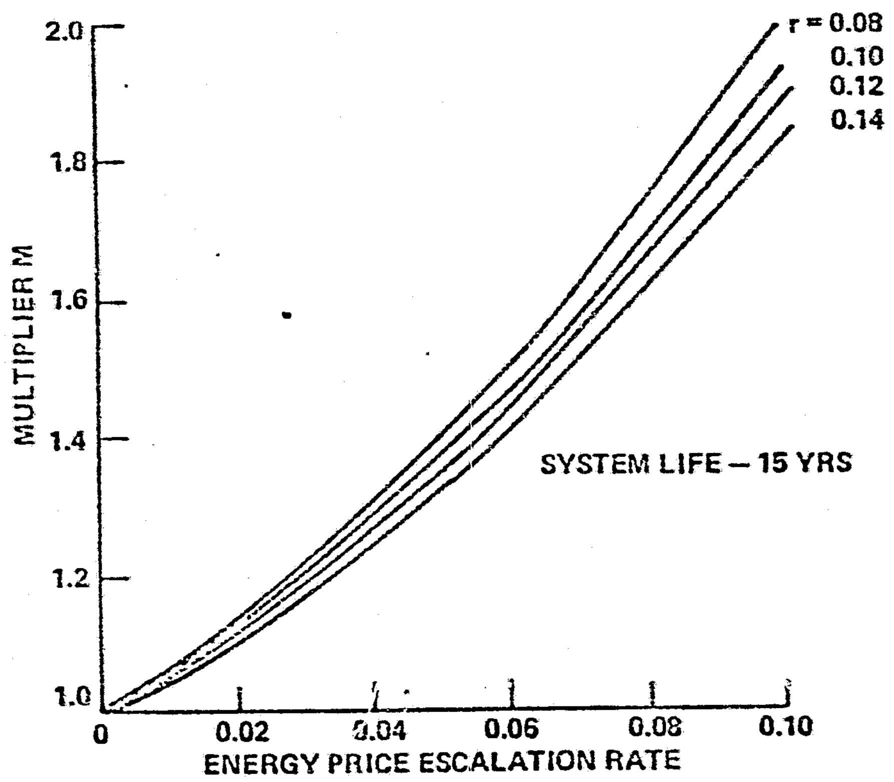


Figure 4.3-5. Levelized Annual Savings = Levelized Annual Benefits - Levelized Annual Costs

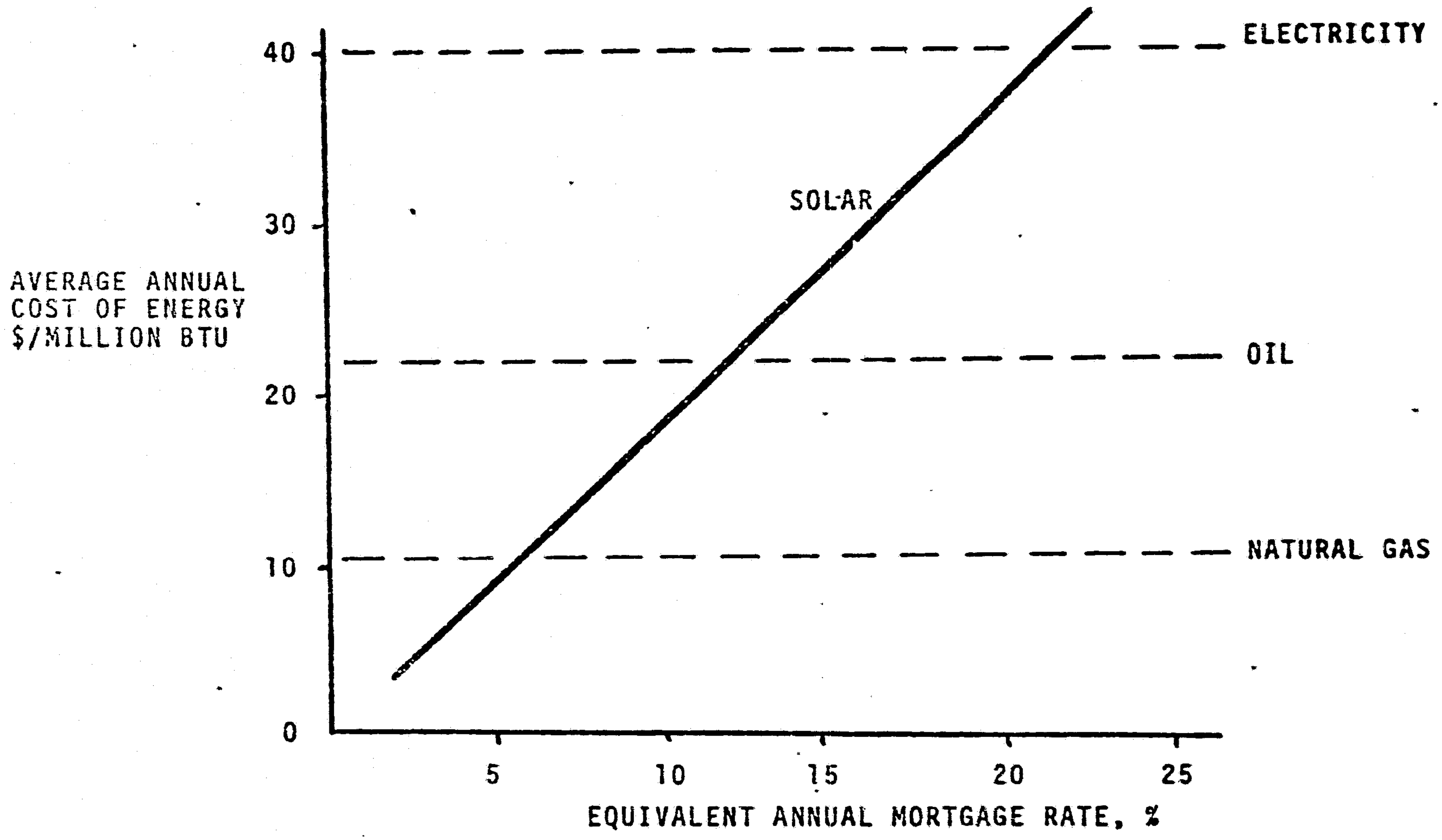


Figure 4.3-6. Comparing Solar and Non-Solar Energy Costs considering 15 Years of Energy Inflation

availability of low cost loans and based on the conventional energy source solar energy is used with. Solar is economic against the use of electric resistance heating in many parts of the country.

The heat pump is a definite saver of electricity over resistance heat. A study was conducted to evaluate the economics of solar heating against an advanced heat pump having a SPF = 3.0 in the Philadelphia area. The results are presented in Figure 4.3-7. Included are curves showing the influence of years to payback based on LAC = LAB over that period and the influence of time of day electric rates. Solar heating is effective as the cost of the installed systems decrease with the early 1980's as the likely time period of occurrence.

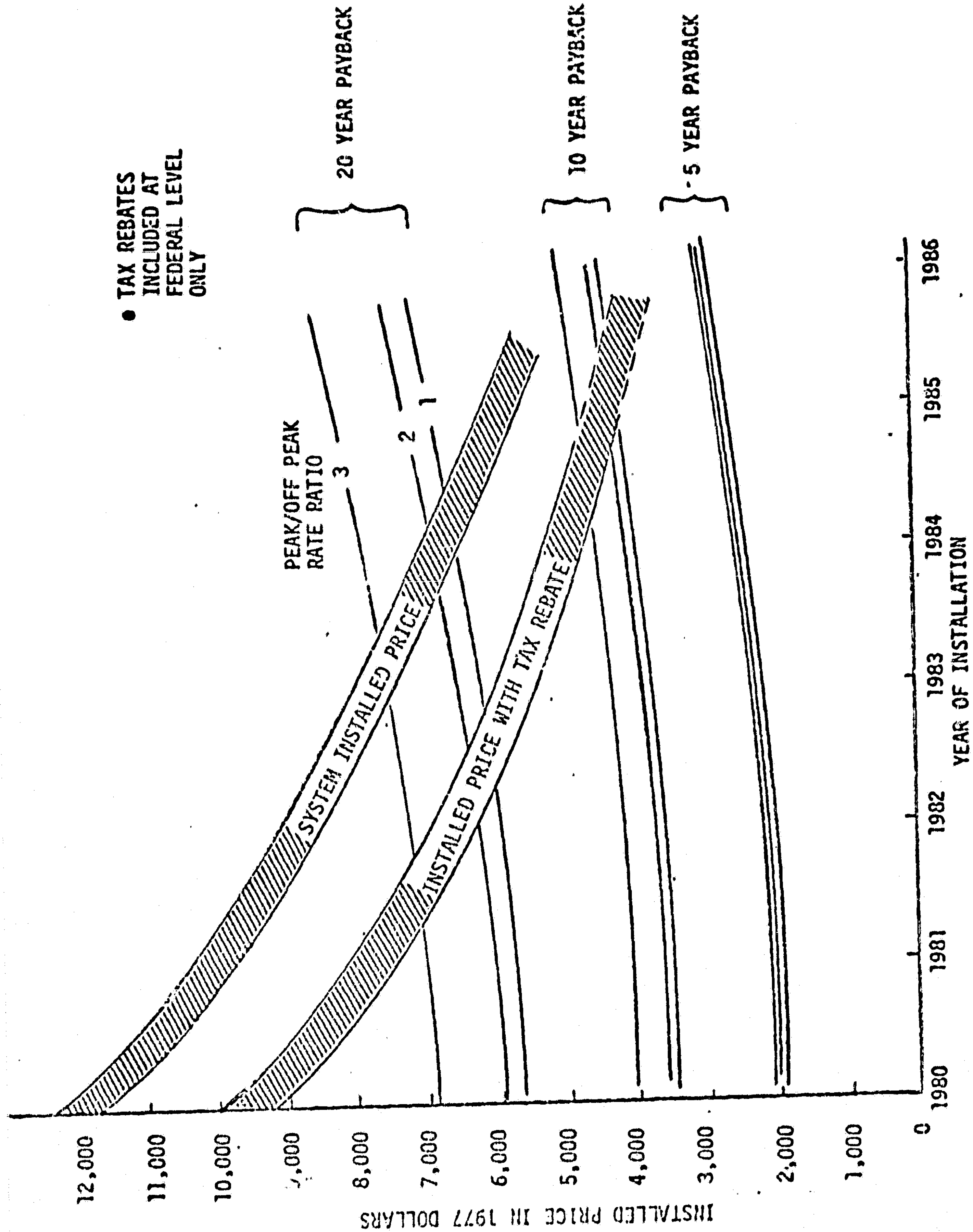


Figure 4.3-7. Economics of Solar Heating and Hot Water System

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4.3.3 Solar Cooling

Solar cooling economics begins with the premise that the collector system, the TES tank, and all other portions of a solar heating and domestic hot water system are in place. Then the energy savings available in the cooling season are compared to the increased cost of the rankine driver heat pump over a conventional unit. When tens of thousands of units are produced, the 3 ton Solartron heat pump is expected to cost the consumer about $3170 compared to $1140 for a conventional unit. Therefore the energy savings accrued during the cooling season must be able to pay for a $2030 increase in investment.

The key measure of the cost of cooling is the energy efficiency rating of the unit (EER), the ratio of cooling BTU's delivered to watts used by the unit. Conventional equipment is getting better with time and the Solartron equipment should benefit from those improvements as well. Figure 4.3-8 shows the estimated EER for both the Solartron equipment and conventional equipment. Typical savings of 1500 KWH/year are expected.

Based upon a 1500 KWH per year savings the effective payback for a 20 year life component varies according to the year of installation and the electrical rate escalation factor. The trends are quantified in Figure 4.3-9. The late 1980's appears to be a reasonable time for which solar cooling can become cost effective.

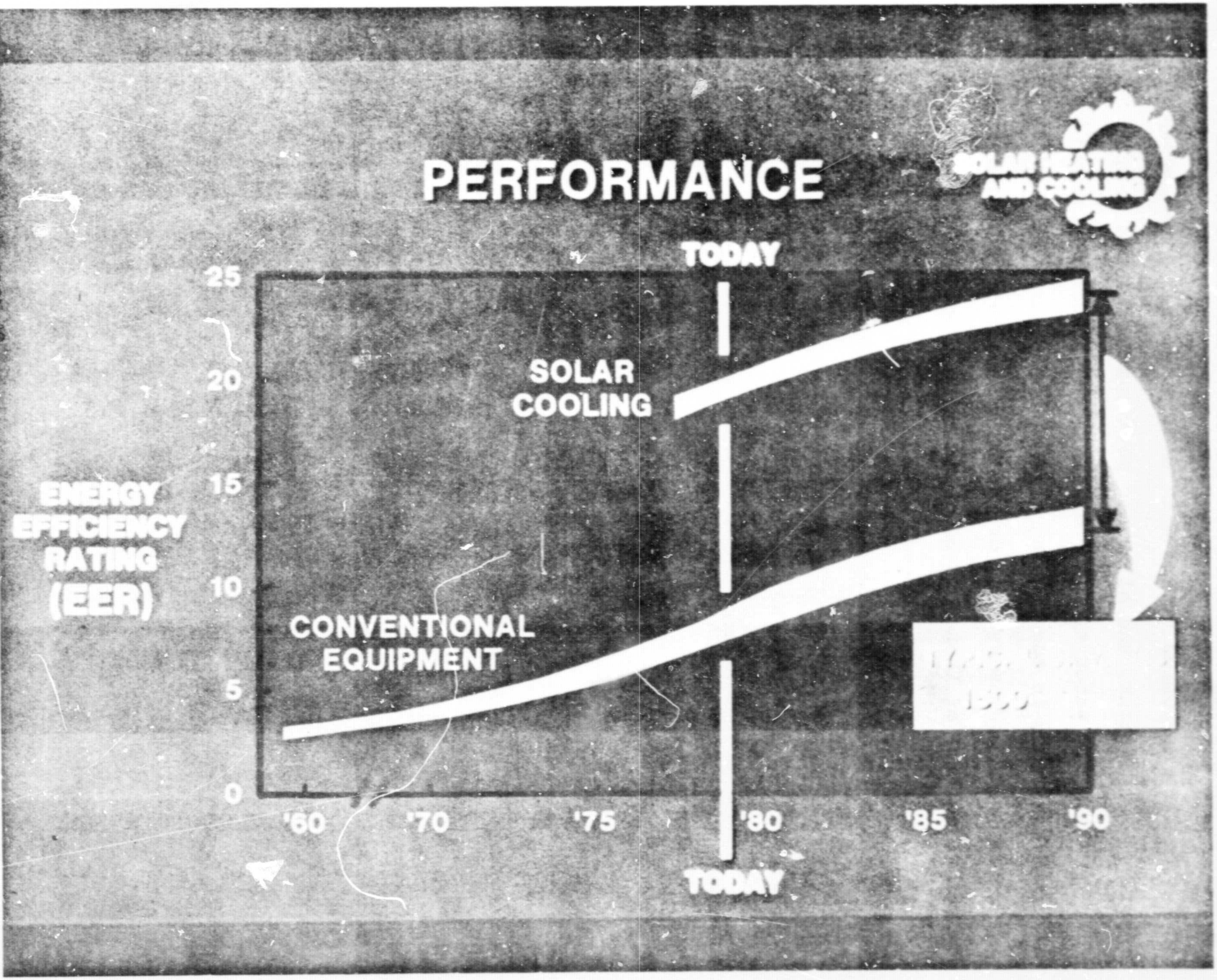

Figure 4.3-8

C-2

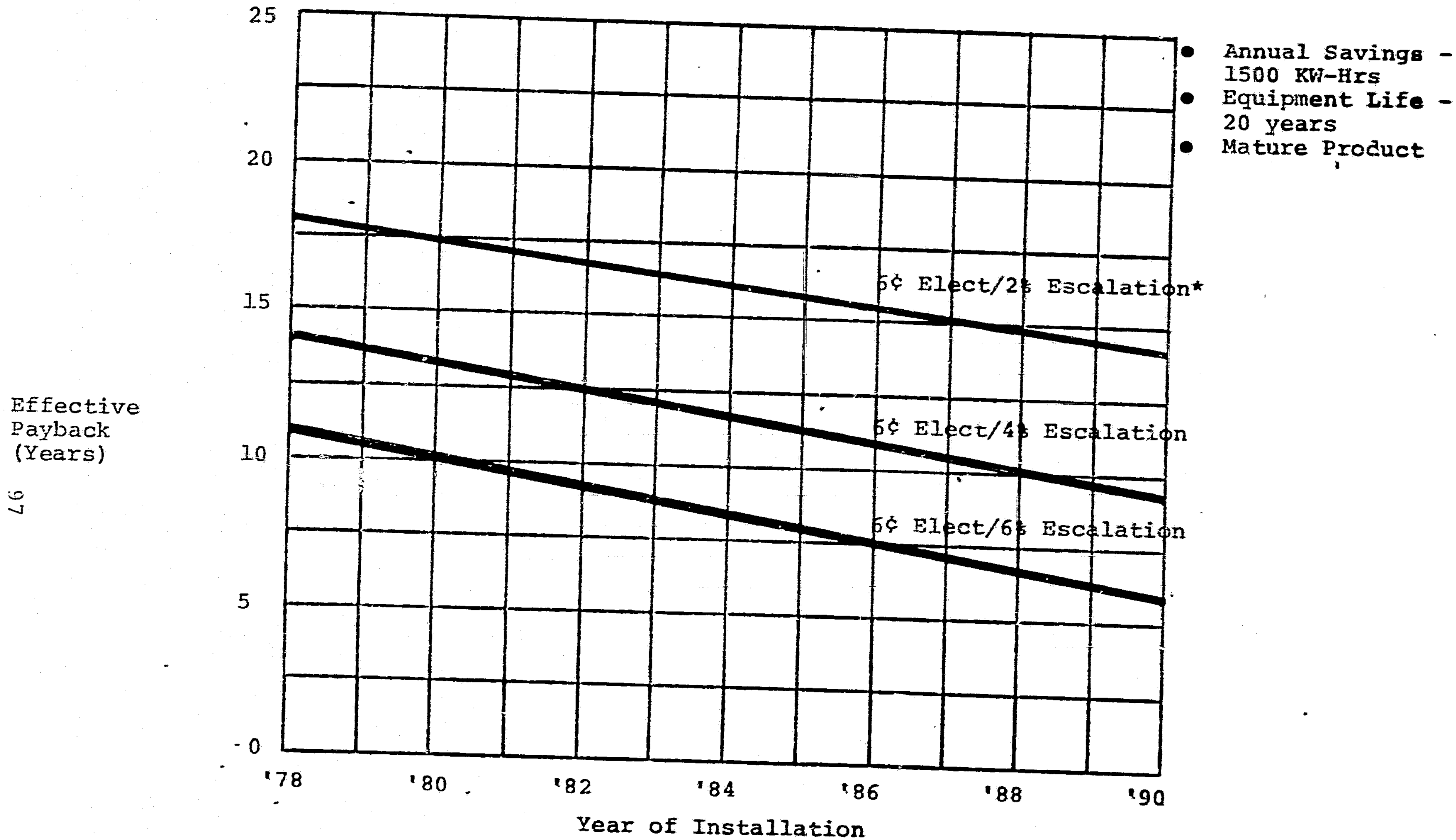

Figure 4.3-9. Cooling Economics

*Above inflation

4.4 COMPARISON OF REH-101 WITH OTHER SOLAR RANKINE COOLING SUBSYSTEMS[1]

As pointed out in the above reference it is difficult to make a direct comparison of the various Rankine driven air conditioners now under development since none of the systems have been tested under the exact same ambient conditions. The usual criteria, EER and $OCOP_{TH}$, can be achieved for a wide variety of collector temperatures, condensing temperatures, evaporator temperatures, compressor efficiency, etc. Also the off-design performance of the various types of Rankine engine/vapor compressors can be quite different. EER for the solar is not well published as is evident in the above reference, but this is a most important parameter to be considered for solar air conditioners. Bob Barber in the reference indicated that parasitic power can well approach that for our all electric system for some solar air conditioners under development. As the previous data shows, the REH-30 and REH-100/1 have a quite favorable EER for thermal conditions near design point conditions.

Table 4.4-I lists performance data for several solar Rankine syst including that for the REH-101. Items 1 through 5 were taken fro the above reference. It should be emphasized that REH-101 is an <u>air cooled</u> system as compared to a <u>water cooled</u> system. It shoul also be noted that REH-101 has design point conditions of 50°F evaporator temperature compared to 45°F for the other systems. In addition, REH-101 has design point solar heated fluid temperat of 300°F compared to 190 to 300°F for the other systems.

1. Barber, R.E., "Solar Rankine Air Conditioning Systems", 1979 Annual ASHRAE Meeting, Detroit, June 24-28, 1979.

Items 6 of Table 4.4-I show test data for the REH-101. In an attempt to present a more meaningful comparison with the other listed systems, the REH-101 data will be normalized to show combined $OCOP_{TH}$ for: (1) 45°F evaporator temperature, (2) 85°F cooling source temperature, (3) 45°F evaporator, 85°F cooling source and 210°F solar fluid temperature. Their normalized values are items 7 through 11 on Table 4.4-I. As indicated above, EER must also be compared to complete the evaluation for solar cooling.

Table 4.4-I

Performance Data of Rankine Solar Cooling Systems

Items	Cooling Capacity, Tons	Rankine Cycle Eff.	$OCOP_{TH}$	Cooling Source Temp., ^{o}F	Evap. or Chilled Water, ^{o}F	Collector Transport Fluid, ^{o}F	
1	6	-	0.55	80 Water	45	210	
2	60	-	0.72	75 Water	45	190	
3	3	.083	0.45	85 Water	45	200	
4	23	.080	0.44	85 Water	45	195	
5	100	.156	0.88	85 Water	45	300	
				REH-101 Performance Data			
6	10	.138	.72	95 Air	50	285	(Test Data)
7	10	.138	.66	95 Air	45	285	Normalized Test Data
8	10	.148	.89	85 Air	50	285	
9	10	.148	.81	85 Air	45	285	
10	10	.148	1.0	85 Air	55	285	
11	10	.10	.55	85 Air	45	210	

5.0 LESSONS LEARNED

The experiences of this development effort have led identification of problems to be avoided, hardware improvements, and approaches that were successful. The successes of the program included:

1) heating system hardware is working

2) Collector Loop II is successful and has simplified the design by using fewer and less expensive components, and also uses a simpler control scheme.

3) The EMM is an attractive concept for residential applications and for the most part simplifies system installation considerably.

4) Temperatures required for cooling can be achieved in the summer.

5) Bidders' briefings resulted in lower construction bids from interested contractors.

There have been a considerable number of problems associated with changes and lack of definition of the instrumentation system which increased the installation costs on the earlier sites. Also, hardware deliveries from vendors have slipped from promised schedules which have had an impact on site installation schedules and costs as well.

One major lesson has surfaced from site installation. Installing contractors require a training program to successfully complete an installation. Specifically, residential HVAC dealers seldom possess all skills required for a solar system installation.

Commercial installations were designed by a mechanical engineering firm and installed by large mechanical contractors. There have been fewer problems encountered with these systems than with residential systems. However, an architect was used for the first residential site but his work offered no simplification of the installation.

Domestic hot water contributions by solar systems can be enhanced by using a recirculation system between the primary thermal storage system and a hot water storage tank. The approach used in this program is effective only for large draws of water but requires less equipment. The trade off between benefits achieved versus capital costs and pump operating costs for a recirculating system needs to be identified.

After installing the Dallas site, it was learned that a significant amount of thermal siphoning occurred when the TES tank was operating above 250°F and with the hydronic coil in the air handler placed in the attic above the TES tank. A check valve was installed but did not alleviate the thermal siphoning condition. The problem was solved by a change in the controls to keep valve V4 in cooling mode until the "heat" switch is activated on the thermostat.

The Normal and Spokane sites demonstrated that the collectors do not readily clean themselves of snow in high snow, very cold climates. At Valley Forge there was no indication that such a problem existed. As a result, covers will be recommended for "snow belt" installations.

A review of the data from the sites showed that the 35 BTU/hr ft^2 of insolation set point on the solar controller allowed some loss of energy from the TES tank under cold ambient conditions. A redesigned solar controller now offers a change of set point by the addition of a resistor to accommodate local installation conditions.

Most problems with equipment and installation can be reduced significantly by installer training and owner education.

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